AD-A136 185 PRELIMINARY DESIGN OF AN ALTERNATIVE FUELS COMBINED CYCLE PROPULSION PLANT FOR NAVAL SHIP APPLICATIONS(U) NAVAL POSTGRADUATE SCHOOL MONTEREY CA K S BURGAMY 1/1

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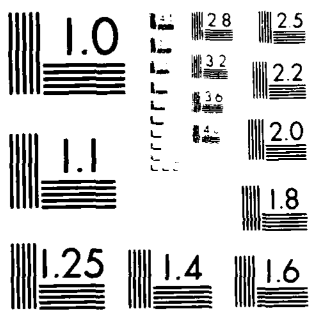

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AD-A136185

REPORT DOCUMENTATION PAGE

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1. REPORT NUMBER	
2. GOVT ACCESSION NO.	AD-A136 185
3. RECIPIENT'S CATALOG NUMBER	
4. TITLE (and Subtitle)	PRELIMINARY DESIGN OF AN ALTERNATIVE FUELS COMBINED CYCLE PROPULSION PLANT FOR NAVAL SHIP APPLICATIONS
5. TYPE OF REPORT & PERIOD COVERED	THESIS
6. PERFORMING ORG. REPORT NUMBER	
7. AUTHOR(s)	BURGAMY, KIRK S.
8. CONTRACT OR GRANT NUMBER(s)	
9. PERFORMING ORGANIZATION NAME AND ADDRESS	MASS. INST. OF TECHNOLOGY CAMBRIDGE, MA 02139
10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS	
11. CONTROLLING OFFICE NAME AND ADDRESS	CODE 031 NAVAL POSTGRADUATE SCHOOL MONTEREY, CA 93940
12. REPORT DATE	JUN 81
13. NUMBER OF PAGES	88
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)	
15. SECURITY CLASS. (of this report)	UNCLAS
15a. DECLASSIFICATION/DOWNGRADING SCHEDULE	
16. DISTRIBUTION STATEMENT (of this Report)	APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)	
18. SUPPLEMENTARY NOTES	
19. KEY WORDS (Continue on reverse side if necessary and identify by block number)	PROPULSION DESIGN ALTERNATIVE FUELS PROPULSION
20. ABSTRACT (Continue on reverse side if necessary and identify by block number)	ATTACHED

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S/N 0102-014-6601

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83 12 21 010

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The results show that a maximum thermodynamic efficiency (η_t) is achieved in the range of π_c = 5 to 6. Total volume on the other hand has a minimum around π_c = 3.5 and climbs steeply above that. A trade-off exists between efficiency and volume with a good design range of π_c = 4 to 5. The design point was ultimately chosen at π_c = 4.5 with a η_t = .48.

The design of a modern coal burning combined cycle offering high efficiency at both rated and part load is within the reach of present technology. Heat exchangers and turbomachinery design are well within the state of the art and PFB technology is expected to come on line within the next few years.

DD Form 1473 1 Jan 73
S/N 0102-014-6601

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Preliminary Design of
An Alternative Fuels Combined Cycle Propulsion Plant
for Naval Ship Applications

by

Kirk Steven Burgamy

B.S. in Naval Arch., U.S. Naval Academy
(1973)

SUBMITTED TO THE DEPARTMENT OF
OCEAN ENGINEERING IN PARTIAL
FULFILLMENT OF THE REQUIREMENTS
FOR THE DEGREES OF

OCEAN ENGINEER

and

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

May 1981

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PRELIMINARY DESIGN OF AN

ALTERNATIVES FUELS COMBINED CYCLE PROPULSION PLANT

FOR NAVAL SHIP APPLICATIONS

by

KIRK STEVEN BURGAMY

Submitted to the Department of Ocean Engineering
on May 1, 1981 in partial fulfillment
of the requirements for the Degrees of Ocean
Engineer and Master of Science in
Mechanical Engineering

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A preliminary design of a coal burning 25,000 HP naval ship propulsion plant was completed. The system consists of a combined Closed Brayton cycle and bottoming Rankine cycle. Coal combustion capability is achieved by using a Pressurized Fluidized Bed (PFB) heater. A thermodynamic analysis and volume estimate was conducted for various Brayton cycle compressor pressure ratios (π_c).

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Thesis Supervisor: A. Douglas Carmichael

Title: Professor of Power Engineering

ACKNOWLEDGEMENTS

The author wishes to thank Professor A. Douglas Carmichael for his guidance, recommendations, and helpful discussions during the preparation of this thesis.

In addition, I would like to thank my wife and typist, Sandy for her understanding and moral support not only in this work, but throughout my three years of graduate study.

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

I. INTRODUCTION

1. Purpose. Increasing prices and decreasing supplies of fuel oil has created a need for highly efficient ship propulsion systems capable of utilizing fuels other than oil. The purpose of this thesis is to perform a preliminary design of a combined cycle propulsion plant equipped with a multi-fuels combustor capable of burning coal or oil.

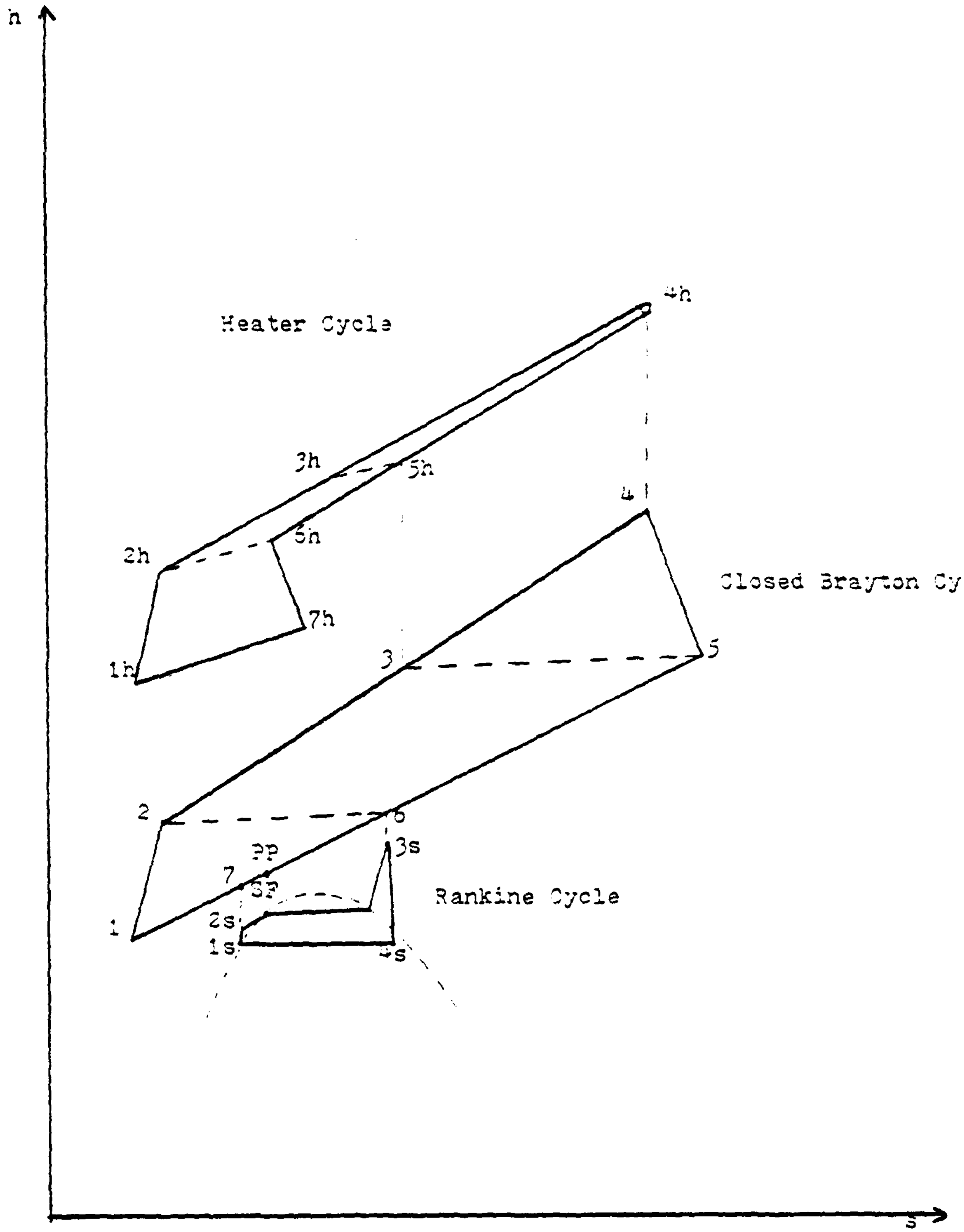
2. System Description. The system investigated is a closed regenerative Brayton Cycle combined with a bottoming Rankine cycle. Working fluids for the cycles are air and water respectively. The heater for the Brayton cycle is a pressurized fluidized bed (PFB). The heater cycle is essentially an open regenerative Brayton cycle. This thesis was conducted considering the heater only in its coal burning mode of operation. Work done by the Brayton and Rankine cycles will drive a common shaft through a double reduction type gear. An enthalpy-enthropy diagram is shown on Figure 1.

3. Scope.

a. Objectives. The objectives of this thesis are to:

(1) Perform a thermodynamic analysis of the system for various closed Brayton cycle pressure ratios utilizing

Figure 1.


h
4h
Heater Cycle
3h
5h
4
6h
2h
Closed Brayton Cycle
7h
1h
3
5
2
6
3s
PP
7
SP
Rankine Cycle
2s
1
1s
4s
s

realistic values for component efficiencies and losses.

(2) Estimate major component volumes and thus obtain a total volume for set of parameters listed in (1) above.

(3) Select a set of parameters for design based upon the above results.

(4) Discuss the technical feasibility and implications of the design as a means for naval ship propulsion.

b. Design Criteria.

(1) The system parameters selected for design were those which gave the lowest total volume within an acceptable range of thermal efficiency.

(2) An acceptable range of thermal efficiency is defined as ± 1% of the maximum efficiency.

c. Limitations to Scope.

(1) No component or system testing was carried out.

(2) The system was designed with the aid of a computer utilizing performance parameters of currently available equipment.

(3) The concept of pressurized fluidized bed (PFB) combustion has not been proven in a shipboard environment. The thesis was conducted based on the assumption that a PFB system would be put to sea in the near future.

II. THERMODYNAMIC ANALYSIS

1. The Brayton Cycle

a. Working cycle. This is a closed Brayton cycle utilizing air as the working fluid. Air was chosen since this is proposed for surface ship application and would thus eliminate the need for carrying a reserve supply of gas if something like Helium or CO_2 were used.

Advantages of a closed cycle over an open cycle are the following:

(1) Compressor suction pressure is no longer limi c to ambient. By increasing the working pressure of the cyc a significant reduction in engine size can be achieved for specified output.

(2) Higher working pressures also increases the heat transfer capacity of the working fluid thus resulting in smaller, more effective heat exchangers.

(3) The use of a high pressure fluid gives the added advantage of regulating the cycle pressure level by varying the inventory of gas in the loop. If this is done in a manner which kept the maximum temperature and speed constant, the turbomachinery vector diagrams would remain essentially unchanged. The result is little or no change in compressor or turbine efficiencies with varying load, thus maintaining high part load efficiency.

A Temperature - entropy (T-s) diagram of the working cycle is shown in Figure 2.

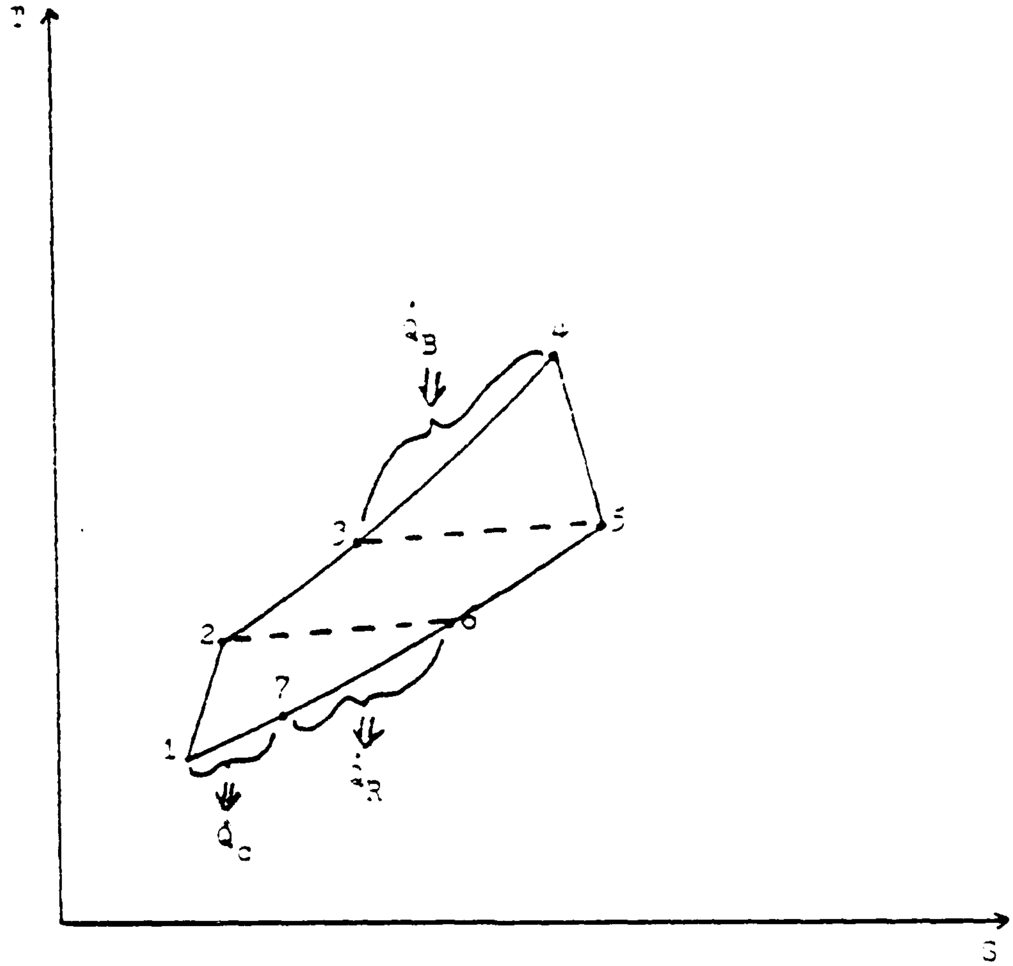


Figure 2.

point 1: compressor suction, cooler discharge
2: compressor discharge, regen inlet (cold)
3: regen outlet (cold), heater inlet
4: heater outlet, turbine inlet
5: turbine outlet, regen inlet (hot)
6: regen outlet (hot), WHB inlet
7: WHB outlet, cooler inlet

$\dot{Q}_B$: heat input from heater cycle

$\dot{Q}_R$: heat released to Rankine cycle

$\dot{Q}_c$: heat released to sea water cooler

The analysis was conducted by assuming values for T_1, T_4, $\eta_{c(p)}$ (compressor polytropic efficiency), $\eta_{T(P)}$ (turbine polytropic efficiency), E_R (regenerator effectiveness), and $\left(\frac{\Delta P}{P}\right)_T$ (total pressure drop).

The specific heat was assumed to vary with temperature according to the following relation:

$$Cp(T) = .2475 - (3.759x10^{-5})\ T - (5.106x10^{-8})T^2 - (1.231x10^{-11})T^3$$

for $\quad$ T in ^oR

Cp in BTU/lbm-^oR

<u>The following are defined:</u>

π_c	-	compressor pressure ratio (P_2/P_1)
π_t	-	turbine pressure ratio (P_5/P_4)
$\left(\frac{\Delta P}{P}\right)_T$	-	% total pressure drop around closed Brayton loop
$\dot{W}_B$	-	power (net) from closed Brayton cycle
$\dot{m}_B$	-	mass flow of working fluid in Brayton cycle
T_i	-	temp in ^oR at point i
h_i	-	enthalpy at point i
R	-	gas constant
P_i	-	pressure abs. at point i
v	-	specific volume
E_R	-	regenerator effectiveness

The basic procedure was to determine T_2, T_3, T_5, T_6, $\dot{Q}_{in}$, and $\dot{W}_B$ for various pressure ratios (π_c).

Determination of T_2 (ref. Dixon):

Small stage polytropic efficiency is

$$\eta_c(p) = \frac{dh_{is}}{dh} = \frac{v\,dp}{C_p\,dT}.$$

since for an isentropic process $T_{ds} = 0 = dh_{is} - v\,dp$

Substituting $v = RT/p$ gives

$$\eta_c(p) = \frac{R}{C_p} \cdot \frac{T}{p} \cdot \frac{dp}{dT}$$

thus,

$$\int_{T_1}^{T_2} \frac{C_p}{T}\,dT = \frac{R}{\eta_c(p)} \int_{p_1}^{p_2} \frac{dp}{p} = \frac{R}{\eta_c(p)} \ln\left(\frac{p_2}{p_1}\right)$$

By substituting the equation for $C_p = C_p(T)$ one gets

$$.2475 \ln\left(\frac{T_2}{T_1}\right) - 3.759\times10^{-5}(T_2 - T_1) + (2.553\times10^{-8})(T_2^{\,2} - T_1^{\,2}) -$$

$$(4.103\times10^{-12})\,(T_2^{\,3} - T_1^{\,3})$$

$$= \frac{R}{\eta_c(p)} \ln \pi_c$$

T_2 can then be determined iteratively.

Determination of T_5:

In a manner similar to the compressor above,

$$\int_{T_4}^{T_5} \frac{C_p}{T}\,dT = R\,\eta_t(p) \ln\left(\frac{p_5}{p_4}\right)$$

After integrating, this becomes

$$.2475 \ln\left(\frac{T_5}{T_4}\right) - (3.759\times10^{-5})(T_5-T_4) + (2.553\times10^{-8})(T_5^{2}-T_4^{2}) -$$

$$(4.103\times10^{-12})\ (T_5^{3} - T_4^{3})$$

$$= R\,\eta_{t(p)} \ln(\pi_t)$$

T_5 can then be determined iteratively.

<u>Determination of T_3 and T_6:</u>

T_3 and T_6 can be found from an assumed regenerator effectiveness (E_R) using the following relation:

$$E_R = \frac{T_5 - T_6}{T_5 - T_2} = \frac{T_3 - T_2}{T_5 - T_2}$$

b. Heater Cycle. This is an open Brayton cycle also using combustion air as the working fluid. The heater itself is a pressurized fluidized bed utilizing coal as fuel. A T-s diagram of the heater cycle is shown in Figure 3.

Major components of the cycle consist of a heater, recuperator and turbine driven compressor. The turbine is powered by exhaust gases leaving the recuperator.

The analysis was conducted by assuming values of T_{1h}, T_{7h}, $\eta_c(P)$, $\eta_T(P)$, and ϕ (equivalence ratio). Both the exhaust products and combustion air were assumed to be perfect gases with specific heats varying in accordance with the relation given in 1.(a).

The following are defined:

π_{ch}	-	heater compressor pressure ratio (P_{2h}/P_{1h})
π_{th}	-	heater turbine pressure ratio (P_{7h}/P_{6h})
$\left(\frac{\Delta P}{P}\right)_{Th}$	-	% total pressure drop around heater loop
$\dot{m}_a$	-	mass flow of air in heater cycle
$\dot{m}_f$	-	mass flow of fuel in heater cycle
$(F/A)_s$	-	stoichiometric fuel - air ratio of fuel used
LHV	-	Lower Heating Value of fuel used
T_{ih}	-	absolute temp at point i_h
h_{ih}	-	enthalpy at point i_h
η_H	-	efficiency of the heater cycle
E_{REC}	-	recuperator effectiveness

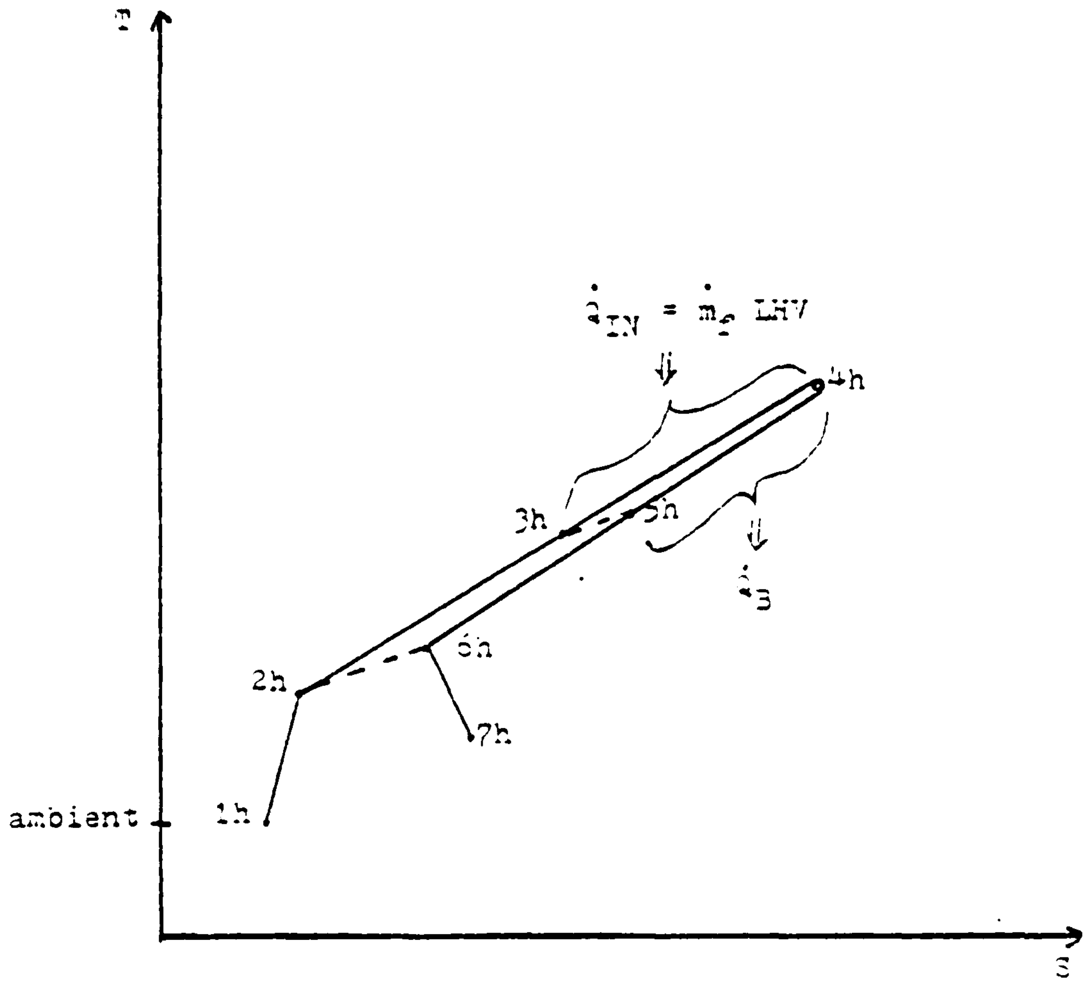


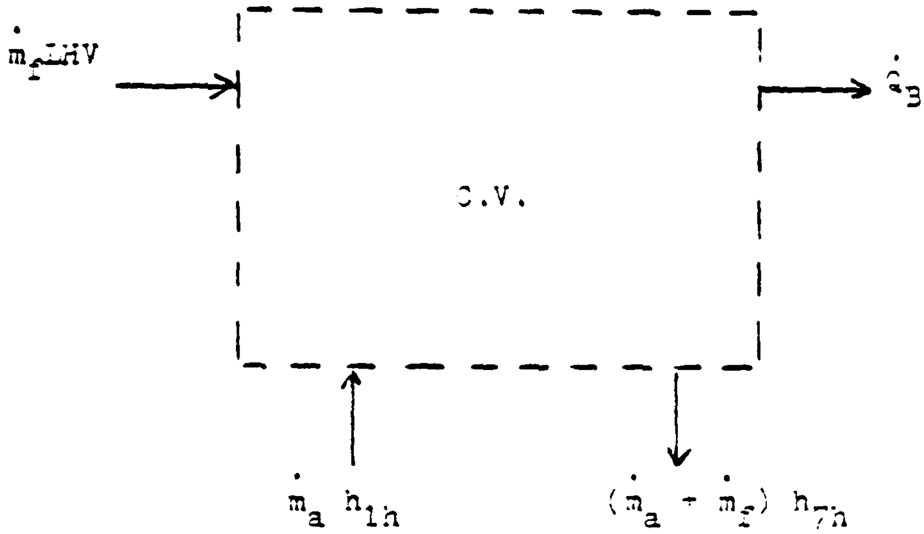
Figure 3.

point 1h: compressor inlet
2h: compressor outlet, recup inlet (cold)
3h: recup outlet (cold), heater inlet
4h: max air temp in heater
5h: heater outlet, recup inlet (hot)
6h: recup outlet (hot), turbine inlet
7h: turbine exhaust

$\dot{Q}_3$: heat released to working cycle

$\dot{Q}_{in}$: heat in = $\dot{m}_f$ LHV

Assume the heater cycle to be a control volume (C.V.) as shown in Figure 4.



<u>Figure 4.</u>

$\dot Q_3$ leaving the C.V. must be equal to $\dot Q_3$ entering the working cycle thus,

$$\dot Q_3 = \dot m_3\ (h_4 - h_3) = (\dot m_a + \dot m_f)(h_{4h} - h_{5h})$$

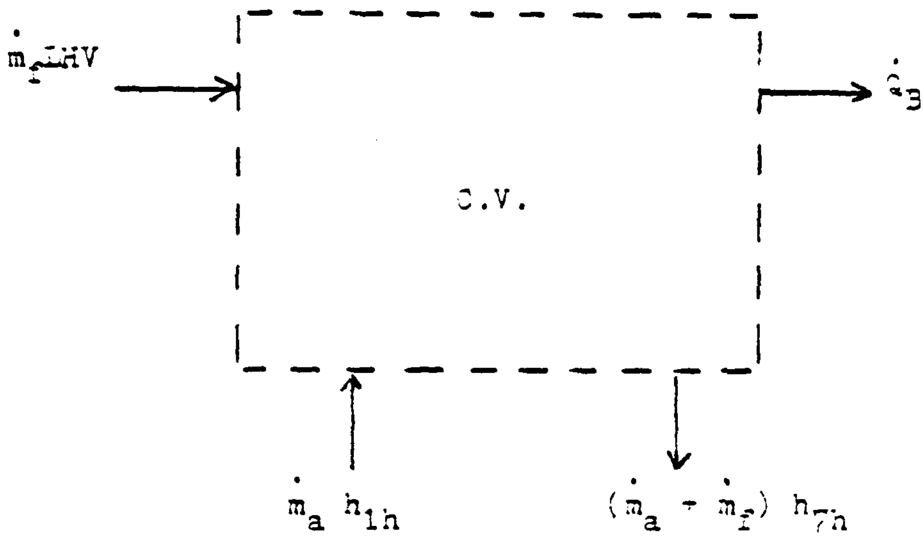
Performing a heat balance on the C.V. gives us

$$\dot Q_3 + (\dot m_a + \dot m_f)\ h_{7h} = \dot m_f(LHV) + \dot m_a h_{1h}.$$

Therefore,

$$\eta_t \equiv \frac{\dot Q_3}{\dot Q_{in}} = \frac{\dot Q_3}{\dot m_f\, LHV} = 1 - \left[\left(1 + (F/A)_s\, \phi\right) h_{7h} - h_{1h}\right] \Big/ \left[(F/A)_s\, \phi\, LHV\right]$$

Assume the heater cycle to be a control volume (C.V.) as shown in Figure 4.



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Performing a heat balance on the C.V. gives us

$$\dot{Q}_3 + (\dot{m}_a + \dot{m}_f) h_{7h} = \dot{m}_f(LHV) + \dot{m}_a h_{1h}.$$

Therefore,

$$\eta_t \equiv \frac{\dot{Q}_3}{\dot{Q}_{in}} = \frac{\dot{Q}_3}{\dot{m}_f LHV} = 1 - \left[\left(1 + (F/A)_s \Phi\right) h_{7h} - h_{1h}\right] \Big/ \left[(F/A)_s \Phi LHV\right]$$

From a power balance between the turbine and compressor

$$\dot{W}_C = \dot{W}_T \text{ (no net work)}$$

$$\dot{m}_a \ \bar{C}_{pc} \ (T_{h2} - T_{h1}) = (\dot{m}_a + \dot{m}_f) \ \bar{C}_{pT} \ (T_{6h} - T_{7h})$$

but

$$T_{2h} = T_{1h} \left(\pi_{ch}\right)^{C_{pc}/\eta_c(p)}$$

$$T_{6h} = T_{7h} \left(\pi_{th}\right)^{C_{pT}\,\eta_t(p)} \qquad \pi_{th} = \left(\left[1 - \left(\frac{\Delta P}{P}\right)_{Th}\right] \pi_{ch}\right)^{-1}$$

thus,

$$\bar{C}_{pc} \, T_{1h} \left[\left(\pi_{ch}\right)^{C_{pc}/\eta_c(p)} - 1\right] = \left[1 + \left(F/A\right)_S \, \phi\right] \bar{C}_{pt} \, T_{7h} \left[\left(\pi_{th}\right)^{C_{pT}\,\eta_t} - 1\right]$$

$$= \left[1 + \left(F/A\right)_S \, \phi\right] \bar{C}_{pt} \, T_{7h} \left\{\left[1 - \left(\frac{\Delta P}{P}\right)_{Th}\right]^{C_{pt}\,\eta_t(p)} \left[\left(\pi_c\right)^{C_{pT}\,\eta_t} - 1\right]\right\}$$

or,

$$\bar{C}_{pc} \, T_{1h} \left[\left(\pi_{ch}\right)^{C_{pc}/\eta_c} - 1\right] + \left[1 + \left(F/A\right)_S \, \phi\right] \bar{C}_{pt} \, T_{7h} =$$

$$\left[1 + \left(F/A\right)_S \, \phi\right] \bar{C}_{pt} \, T_{7h} \left[1 - \left(\frac{\Delta P}{P}\right)_{Th}\right]^{C_{pt}\,\eta_t} \pi_c^{\,C_{pt}\,\eta_t}$$

therefore,

$$\left[1-\left(\frac{\Delta P}{P}\right)_{Th}\right]^{c_p \eta_t} = \frac{\bar{C}_{pc}\, T_{1h}\left[(\pi_{ch})^{c_p/\eta_c}-1\right]+\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\, T_{7h}}{\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\, T_{7h}\,(\pi_{ch})^{c_p \eta_t}}$$

For minimum volume heat exchangers we want the largest $(\Delta P/P)_{Th}$ compatible with the above eqn.

differentiating,

$$c_p\eta_t\left[1-\left(\frac{\Delta P}{P}\right)_{Th}\right]^{c_p\eta_t-1}\left(\frac{-\partial\left(\frac{\Delta P}{P}\right)_{Th}}{\partial\pi_{ch}}\right) =$$

$$\frac{\left\{\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\,T_{7h}\,(\pi_{ch})^{c_p\eta_t}\,\bar{C}_{pc}\,T_{1h}\,\bar{C}_{pc}/\eta_c\,(\pi_{ch})^{c_p/\eta_c-1}\right\}-\left\{\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\,T_{7h}\,\bar{C}_{pt}\,\eta_t\,(\pi_{ch})^{c_p\eta_t-1}\,\bar{C}_{pc}\,T_{1h}\left(\bar{C}_{pc}\,T_{1h}\left[\pi_{ch}{}^{c_p/\eta_c}-1\right]+\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\,T_{7h}\right)\right\}}{\left\{\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\,T_{7h}\,(\pi_{ch})^{c_p\eta_t}\right\}^2}$$

Setting $\dfrac{\left(\frac{\Delta P}{P}\right)_{Th}}{\pi_{ch}} = 0$ and solving for π_{ch} gives

$$\pi_{ch} = \left\{\frac{\left[1+(F/A)_s\,\phi\right]\bar{C}_{pt}\,\eta_t\left(\bar{C}_{pt}\,T_{7h}-\bar{C}_{pc}\,T_{1h}\right)}{\bar{C}_{pc}\,T_{1h}\left(\bar{C}_{pc}/\eta_c-\bar{C}_{pt}\,\eta_t\right)}\right\}^{\eta_c/c_p}$$

By substitution

$$\left(\frac{\Delta P}{P}\right)_{Th} = 1 - \left\{ \frac{\left[1+(F/A)_s \phi\right] \bar{C}_{pt} T_{7h} + \bar{C}_{p_c} T_{1h} \left[(\Pi_{ch})^{C_p/\gamma_c} - 1\right]}{\left[1+(F/A)_s \phi\right] \bar{C}_{pt} T_{7h} (\Pi_{ch})^{C_p/\gamma_t}} \right\}^{\frac{1}{C_p/\gamma_t}}$$

Thus Π_{ch} and $\left(\frac{\Delta P}{P}\right)_{Th}$ can be found through known or assumed quantities.

Determination of T_{3h}, T_{5h}

$$E_{rec} \equiv \frac{C_h (T_{5h} - T_{6h})}{C_{min} (T_{5h} - T_{2h})} = \frac{C_c (T_{3h} - T_{2h})}{C_{min} (T_{5h} - T_{2h})}$$

where $$C_h = \bar{C}_{p_h} (\dot{m}_a + \dot{m}_f)$$

$$C_c = \bar{C}_{p_c} \dot{m}_a$$

therefore,

$$T_{3h} = T_{2h} + \left[1 + (F/A)_s \phi\right] (T_{5h} - T_{6h})$$

T_{2h} and T_{6h} can be found once Π_{ch} is determined. Assume that

$$T_{5h} = T_3 + 100^\circ R$$

2. The Rankine Cycle. This is a bottoming steam cycle that recovers heat from the closed Brayton cycle that would otherwise be discharged overboard through the cooler. The relatively hot gas leaving the regenerator is pre-cooled in the Waste Heat Boiler (WHB) before being sent to the cooler which brings the temperature down to compressor inlet conditions. A T-s diagram of the Rankine cycle is shown in Figure 5.

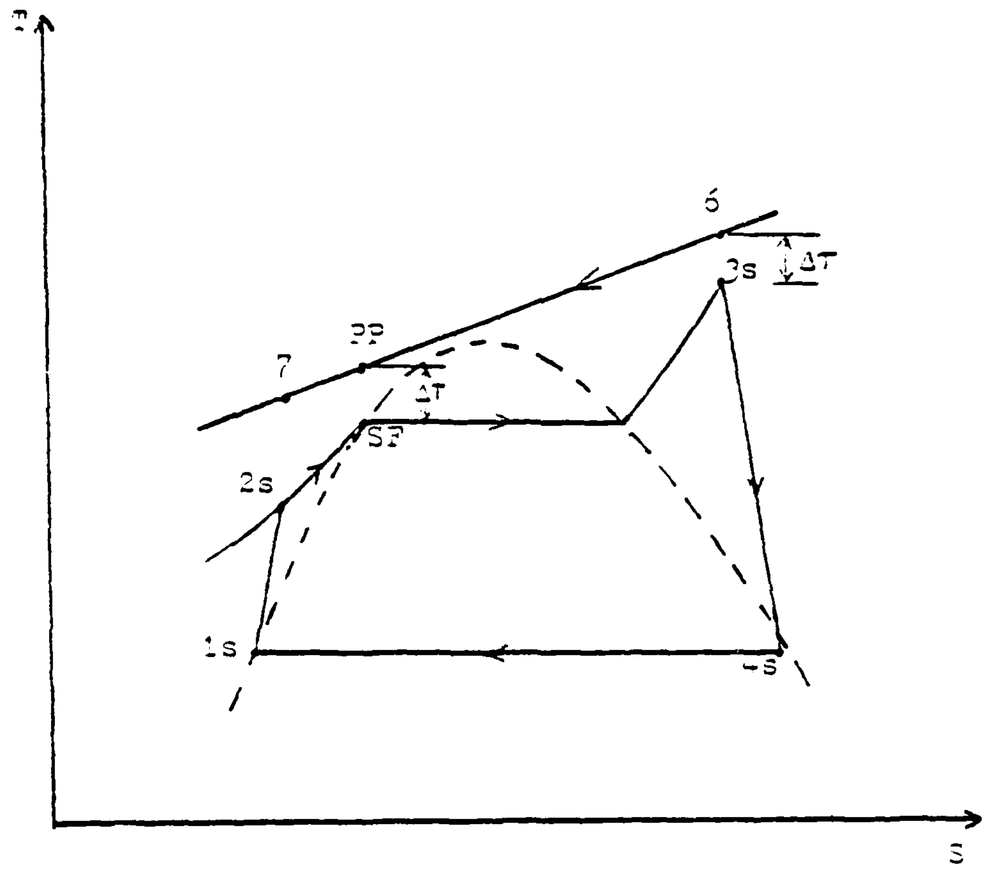


Figure 5.

Major components of the cycle are the WHB, condenser, steam turbine, and feed pump. It is assumed that the feed pump is driven with power from the turbine.

The analysis was conducted by assuming values of T_{1s}, ΔT, $\eta_{p}(is)$ (pump isentropic efficiency), $\eta_{T}(is)$ (turbine isentropic efficiency), and η WHB (waste heat boiler efficiency).

<u>The following are defined</u>:

h_{is}	-	enthalpy at point is
T_{is}	-	absolute temp at point is
PP	-	pinch point on Brayton side
SP	-	saturation point on steam side
β	-	mass flow ratio ($\dot{m}_S/\dot{m}_B$)
x	-	quality of steam exiting turbine
P_{is}	-	absolute pressure at point is
$\dot{W}_R$	-	power (net) from the steam cycle
η_{t}	-	thermal efficiency of the combined cycle
$\dot{m}_S$	-	mass flow of steam in Rankine cycle

Rankine cycle parameters were chosen for maximum work. The procedure was as follows:

(1) <u>Compute T_{3S}</u> from the relation

$$T_{3S} = T_6 - \Delta T$$

(2) <u>Assume a value for T_{pp}</u>. $T_1 < T_{pp} < 662.0^{\circ}F$

(3) Compute T_{SF} from the relation

$$T_{SF} = T_{pp} - \Delta T$$

(4) Obtain P_{SAT}, h_{SF}, h_{3S} from steam tables.

(5) Compute h_{4S} from the relation

$$h_{4S} = h_{3S} - \eta_t (is)(h_{3S} - h_{4S}(is))$$ get $h_{4S}(is)$ from steam tabl[illegible]

(6) Obtain x (turbine exit), h_{1S} from steam tables

(7) Compute h_{2S} from the relation

$$h_{2S} = h_{1S} + \frac{h_{2S}(is) - h_{1S}}{\eta_P (is)}$$

(8) Compute β from the relation

$$\beta = \frac{\dot{m}_s}{\dot{m}_3} = \frac{\eta_{WHB}(h_6 - h_{PP})}{h_{3S} - h_{SF}}$$

(9) Compute $\dot{W}_R$ from the relation

$$\dot{W}_R = \dot{m}_3 \beta \left[(h_{3S} - h_{4S}) - (h_{2S} - h_{1S}) \right]$$

*(10) Iterate with T_{pp} (step 2) until $\dot{W}_R$ is maximum.

(11) Compute T_7 from the relation

$$C_p(T_7) \cdot T_7 = C_p(T_{PP}) \cdot T_{PP} - \beta (h_{SF} - h_{2S}) / \eta_{WHB}$$

(12) Calculate η_t from the relation

$$\eta_t = \frac{\dot{W}_B + \dot{W}_R}{\dot{q}_B}$$

III. VOLUME ANALYSIS

1. Turbomachinery. An analytical procedure for the minimum volume design of turbomachinery was developed by Lee (ref. 10). The following equations and mathematical expressions summarize that portion of the procedure which are applicable to compressor and turbine designs in this thesis. Volume obtained by Lee's method neglects the volume of casing, inlet and outlet plenums, and bearing assemblys.

a. Turbine. Free vortex blading and axial exit stages were assumed throughout the analysis.

The following are defined:

S.F.	-	factor of safety
U_t	-	max. allowable tip speed
ϕ_t	-	flow coefficient at tip
ν_a	-	hub-tip ratio at inlet
ν_s	-	hub-tip ratio at exit
C_L	-	clearance factor
V_α	-	specific volume at inlet
r_t	-	max tip radius
n	-	polytropic coefficient
K	-	isentropic exponent
ρ_m	-	density of blade material
α	-	stress concentration factor at root
a	-	amplification factor
ϑ	-	correction factor for taper

x_1 - $\dfrac{\text{dist. from tip to 1st principle axis}}{\text{chord}}$

κ_{zz} - normalized moment of inertia

Vol_t - volume of turbine

The volume was computed by assuming values for SF, U_t, ρ_t, ν_α, C_1, K, ρ_m, α, a, x_1, K_{zz}, and θ.

r_t, n, and ν_B are computed from the following relations:

$$r_t = \left(\dot{m}_3 \nu_\alpha / [\rho_t U_t \pi (1 - \nu_\alpha^2)]\right)^{1/2}$$

$$n = \frac{K}{K - (K-1)\eta_{tp}}$$

$$\nu_B = \left[1 - \pi_t^{\,n} (1 - \nu_\alpha^2)\right]^{-1/2}$$

Compute C_1, C_2, C_3, C_4 from the following:

$$C_1 = 0.2(1+\nu)\nu_\alpha^{-1}(1-\nu_\alpha^2) U_t^2 r_t^2 x_1 / \kappa_{zz}$$

$$C_2 = 2(A - B T_\alpha)/(SF \cdot \alpha)$$

$$C_3 = 2 B T_\alpha (1-\nu_\alpha^2)^{n-1}/(SF \cdot \alpha)$$

$$C_4 = -\theta \rho_m U_t^2 / 2$$

where A and B are material dependent constants. The following can then be calculated:

$$N = C_1 \left[(1-\nu)/(1+\nu)\right] D_t^2 + \nu^2$$

$$D = C_2 + C_3(1-\nu^2)^{1-n} + C_4(1-\nu^2)$$

$$DN = C_1\left[-(\nu_t^2+\nu^2)(1+\nu) - 2\nu(1-\nu)(1+\nu) - (1-\nu)^{-1}(-2\nu)(\nu_t^2-\nu^2)\right]$$

$$DD = 2\nu(1-\nu^2)^{-n}(n-1)C_3 - 2\nu C_4$$

The axial chord for a stage is given by b_h. For a large number of stages, summation of b_h for the whole turbine can be approximated by an integral:

$$I_T = \int_{\nu_\alpha}^{\nu_\beta} d(b_h) = \sum_{\alpha}^{\beta} b_h .$$

where

$$d(b_h) = \frac{1}{2}\left(N/D\right)^{-1/2}\left[(DN\cdot D - DD\cdot N)/D^2\right]d\nu$$

This integral can be evaluated numerically thus,

$$Vol_T = 2\pi r_t^2 C_L I_T$$

b. <u>Compressor</u>. An approach similar to that of the turbine was used for the axial compressor. Axial exit stages were assumed in the analysis.

The volume was computed by assuming values for SF, U_t, ϕ_t, ν_α, C_L, K, ρ_m, α , a, x_1, K_{I_1} , θ, and σ_y (blade material yield stress).

r_t, n, and ν_3 are computed from the following relations:

$$r_t = \left(\dot m_3 V_\alpha / \left[\rho_t u_r \pi (1 - \lambda_\alpha^2) \right] \right)^{1/2}$$

$$n = \frac{K}{K - (K-1)/\eta_c(\rho)}$$

$$\nu_3 = \left[1 - \pi_c^{-1/n} (1 - \lambda_\alpha)^2 \right]^{1/2}$$

Compute C_1, C_2, and C_3 from the following:

$$C_1 = .2(\gamma+1)\lambda_\alpha^{-1}(1-\lambda_\alpha^2) u_r^2 r_t^2 \rho_t^2 \lambda_1 / K_{21}$$

$$C_2 = -\theta_1 \rho_m U_t^2/2$$

$$C_3 = 2 T_y/(x - 3F)$$

Then calculate the following:

$$N = C_1 \left[(1-\nu)/(1+\nu) \right] (\rho_t^2 + .49\nu^2)/(\rho_t^2 + .7225\nu^2)$$

$$D = C_3 + C_2(1-\nu^2)$$

$$\begin{aligned} DN = C_1 \Big[& -(\rho_t^2 + .49\nu^2)/(\rho_t^2 + .7225\nu^2)(1+\nu)^{-1} \\ & - (1-\nu)(1+\nu)^{-2}(\rho_t^2 + .49\nu^2)/(\rho_t^2 + .7225\nu^2) \\ & + .98\nu(1-\nu)(1+\nu)^{-1}/(\rho_t^2 + .7225\nu^2) \\ & - 1.445(1-\nu)(1+\nu)^{-1}(\rho_t^2 + .49\nu^2)(\rho_t^2 + .7225\nu^2)^{-2} \Big] \end{aligned}$$

$$DD = -2 C_2 \nu$$

As with the turbine define

$$I_T = \int_{1/\alpha}^{1/s} d(o_h)$$

where

$$d\,o_h = \frac{1}{2}\left(\frac{N}{D}\right)^{-1/2}\left[(D N' D - D D' N)/D^2\right] d\nu$$

This integral can be evaluated numerically thus

$$Vol_c = 2\pi r_T^2 C_L I_T$$

2. Heat Exchangers. The volumes of the Regenerator, Recuperator, Cooler, and Heater were computed using the method of minimum volume design developed by Lee (ref. 10). In essence Lee found that the optimal diameter ratio (D_h/D) for these heat exchangers can be established as a simple relation to the pressure ratio (π_c) of the cycle. The results of the optimally distributed pressure drops ($\frac{\Delta P}{P}$) as reported by Lee were also studied. Parametric equations relating pressure drop to pressure ratio were developed and used in this analysis. No consideration has been made on computing the volume of the shell, headers, or other appendages. Lee's relations were developed from basic heat transfer and pressure drop relationships.

a. Regenerator. The regenerator is assumed to be a counter flow shell and tube heat exchanger. High pressure is inside the tubes as is the case for current design practices.

The following are defined:

$\left(\frac{\Delta P}{P}\right)_{Reg}$ - % pressure drop across the regenerator (both steams)

$\left(\frac{D_h}{D}\right)_{Reg}$ - diameter ratio for regenerator

$\Delta h_{i\,Reg}$ - enthalphy difference across the interior stream of Reg.

D - tube inside diameter

$\overline{\Delta T}$ - log mean temp. difference of hot and cold streams

P_r - Prandlt number

ρ - density

μ - dynamic viscosity

k - thermal conductivity

Values for $\left(\frac{\Delta P}{P}\right)_{Reg}$ and $\left(\frac{D_h}{D}\right)_{Reg}$ can be estimated from the following relations:

$$\left(\frac{\Delta P}{P}\right)_{Reg} = .135 - .0234\,\pi_c \quad \left(\text{for} \left(\frac{\Delta P}{P}\right)_T = .1,\ E_R = .88\right)$$

$$\left(\frac{D_h}{D}\right)_{Reg} = .557 + .0933\,\pi_c$$

Subscript 1 refers to quantities inside the tubes whereas subscript 2 refers to quantities outside the tubes. A_{Reg} can be computed as follows:

$$A_{Reg} = \left(\pi/4\right) \gamma_R \, \zeta_R \, \dot{m}_g \, \Delta h_R^{\,1.41} \left[1 - \left(\frac{D_h}{D}\right)_R\right]^{2.41} \left[\,\cdot\, - \pi_c^{2} \left(\frac{D_h}{D}\right)_R^{-3}\right]^{.41}$$

where

$$\gamma = 17.24 \, k_1^{-1.41} \, \mu_1^{1.23} \, P_{r1}^{-.564} \, \rho_1^{-.41}$$

$$\zeta = P_1^{-.41} \, D^{1.7} \, \overline{\Sigma}^{-.41}$$

The volume can then be computed as follows:

$$Vol_{Reg} = A_{Reg} \left(\frac{\Delta P}{P}\right)_{Reg}^{-0.41}$$

b. <u>Recuperator</u>. The recuperator is similar in design to the regenerator only it refers to the heater cycle.

<u>The following are defined</u>:

$\left(\frac{\Delta P}{P}\right)_{Rec}$	-	% pressure drop across the recuperator (both streams)
$\left(\frac{D_h}{D}\right)_{Rec}$	-	diameter ratio for recuperator
$\left(\frac{\Delta P}{P}\right)_{h\pi_2}$	-	% pressure drop across the heater side (outside tubes)
$\left(\frac{\Delta P}{P}\right)_{PFB}$	-	% pressure drop across the PFB
$\Delta h_{1\,Rec}$	-	Enthalpy difference across the interior stream of recup.

Values for $\left(\frac{\Delta P}{P}\right)_{Rec}$ and $\left(\frac{D_h}{D}\right)_{Rec}$ can be estimated from the following relations

$$\left(\frac{\Delta P}{P}\right)_{Rec} = \left(\frac{\Delta P}{P}\right)_{T_n} - \left(\frac{\Delta P}{P}\right)_{HTr_1} - \left(\frac{\Delta P}{P}\right)_{PF3}$$

$$\left(\frac{D_h}{D}\right)_{Rec} = .65$$

Subscripts 1 and 2 refer to quantities inside and outside the tubes respectively. A_{Rec} is computed as follows:

$$A_{Rec} = \left(\pi/4\right) \tau_F \gamma_F \dot{m}_{iF} \Delta h_{iF}^{.41} \left[1 + \left(\frac{D_h}{D}\right)_{Rec}\right]^{-2.41} \left[1 - \left(\frac{D_h}{D}\right)_{Rec}\right]^{-3\ -.41}$$

where τ and γ are as previously defined.

Then compute the volume

$$Vol_{Rec} = A_{Rec} \left(\frac{\Delta P}{P}\right)_{Rec}^{-0.41}$$

c. Cooler. The cooler is assumed to be a shell and tube heat exchanger with the gas inside the tubes and salt water outside.

The following are defined:

$\left(\frac{\Delta P}{P}\right)_{Clr}$ - % pressure drop across the cooler (air side)

$\left(\frac{D_h}{D}\right)_{Clr}$ - diameter ratio for cooler

$\Delta h_{i\ Clr}$ - enthalpy difference across the interior stream of cooler

v_{WHB_2} - sp. vol. of working fluid at WHB exit

$\left(\frac{\Delta P}{P}\right)_{WHB}$	-	% pressure drop across the WHB (air side)
$\left(\frac{\Delta P}{P}\right)_{HTR}$	-	% pressure drop across the heater (inside tubes)
$\eta_{p(P)}$	-	polytropic cooling water pump efficiency
V_w	-	sp. vol. of cooling water

Values for $\left(\frac{\Delta P}{P}\right)_{Clr}$ and $\left(\frac{D_h}{D}\right)_{Clr}$ are estimated from the following relations:

$$\left(\frac{\Delta P}{P}\right)_{Clr} = \left(\frac{\Delta P}{P}\right)_{T} - \left(\frac{\Delta P}{P}\right)_{WHB} - \left(\frac{\Delta P}{P}\right)_{Reg} - \left(\frac{\Delta P}{P}\right)_{HTR}$$

$$\left(\frac{D_h}{D}\right)_{Clr} = .092\,\pi_c + .076$$

Subscripts 1 and 2 refer to quantities inside and outside the tubes respectively. A_{Clr} is computed as follows:

$$A_{Clr} = \left(\pi/4\right)\pi_c\,\gamma_c\,\dot{m}_3\,\Delta h_{Clr}^{1.41}\left[1-\left(\frac{D_h}{D}\right)_{Clr}\right]\left[1-\alpha\left(\frac{D_h}{D}\right)_{Clr}\right]^{.41}\left[1+C\beta_c\left(\frac{D_h}{D}\right)_{Clr}^{.5}\right]^{.41}$$

where π , γ are as previously defined and

$$\beta = \left(\dot{m}_1/\dot{m}_2\right)^{-1.75}\left(\mu_1/\mu_2\right)^{-.25}\left(\rho_1/\rho_2\right)\left(P_1/P_2\right)$$

$$C = \left(\dot{m}_2/\dot{m}_1\right)\left(\eta_{p(P)}\,\eta_{p(P)}\right)^{-1}\left(V_w/V_{WHB}\right)\left(\rho_2/\rho_1\right)$$

$$\alpha = \left(k_1/k_2\right)\left(\mu_1/\mu_2\right)^{-.8}\left(\dot{m}_1/\dot{m}_2\right)^{.8}\left(P_{r1}/P_{r2}\right)^{.4}$$

The volume is thus

$$Vol_{clr} = A_{clr} \left(\frac{\Delta P}{P}\right)_{clr}^{-.41}$$

d. <u>Heater</u>. The heater is assumed to be a shell and tube heat exchanger with the working gas in the tubes and combustion gas out.

<u>The following are defined:</u>

$\left(\frac{D_h}{D}\right)_{htr}$	-	diameter ratio for heater
$\Delta h,\ htr$	-	enthaphy difference across the interior stream of heater

Values of $\left(\frac{\Delta P}{P}\right)_{htr_1}$, $\left(\frac{\Delta P}{P}\right)_{htr_2}$, and $\left(\frac{D_h}{D}\right)_{htr}$ are estimated from the following relations:

$$\left(\frac{\Delta P}{P}\right)_{htr_2} = .0108\,\pi_c - .01 \qquad \left[\text{for } \left(\frac{\Delta P}{P}\right) = .\text{[illegible]}\right]$$

$$\left(\frac{\Delta P}{P}\right)_{htr_1} = \left(\frac{\Delta P}{P}\right)_{htr_2} \beta^{-1} \left(\frac{D_h}{D}\right)_{htr}^{3}$$

where

$$\beta = \left(\dot{m}_1/\dot{m}_2\right)^{-1.75} \left(\mu_1/\mu_2\right)^{-.25} \left(\rho_1/\rho_2\right) \left(P_1/P_2\right)$$

$$\left(\frac{D_h}{D}\right)_{htr} = .048\,\pi_c + .214$$

Subscripts 1 and 2 refer to quantities inside and outside

the tubes respectively. Compute A_{htr} as follows:

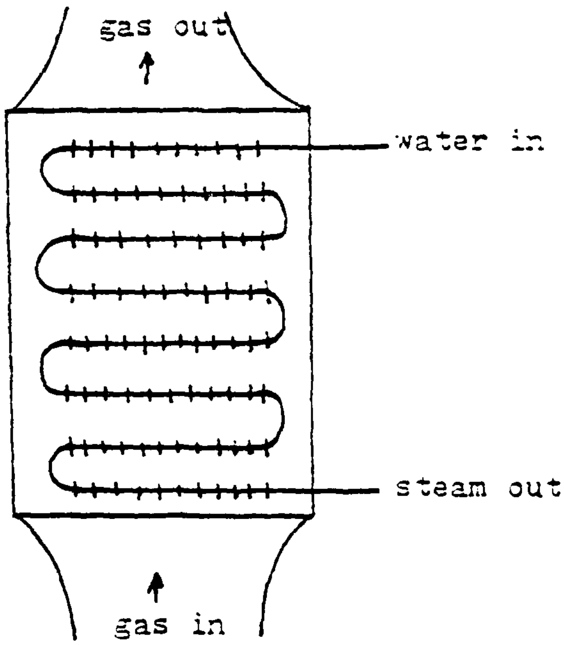
$$A_{htr} = (\pi/4)\,\tau_h\,\gamma_h\,\dot{m}_2\,\Delta n_{htr}^{\,1.44}\left[1 + \left(\frac{D_h}{D}\right)_{htr}\right]\left[1 - \alpha_h\left(\frac{D_o}{D}\right)_{htr}\right]^{-1}\left(\frac{D_h}{D}\right)_{htr}^{-.25}$$

where τ, γ, α are as previously defined.

Then compute Vol_{htr}

$$V_{htr} = A_{htr}\left(\frac{\Delta P}{P}\right)_{htr_2}^{-.41}$$

e. <u>Waste Heat Boiler</u>. The boiler is assumed to be a once through cross-counterflow type, as shown in Figure 6. High pressure is inside the tubes thus saturation pressure must be greater than the working pressure of the closed Brayton cycle.



<u>Figure 6.</u>

Boiler volume is estimated using heat transfer relations and volumes from similar U.S. Navy COGAS (Combined Gas Turbine and Steam) designs (ref. 11).

The following are defined:

- C_{ps} - ave. specific heat on steam side
- C_p - ave. specific heat on air side
- NTU - Number Transfer Units
- #Ps - number of passes
- A_{HT} - Heat transfer area
- U - overall heat transfer coefficient
- V_T - total volume

From a DTNSRDC (David Taylor Naval Ship Research and Development Center) study (ref.11) the following relations are obtained:

$$CR = \frac{CMIN}{CMAX} = \frac{\dot{m}_s C_{ps}}{\dot{m}_a C_p} \quad \text{or} \quad \frac{\dot{m}_a C_p}{\dot{m}_s C_{ps}}$$

$$ER = (\eta_{NHB}\,CR - 1)/(\eta_{NHB} - 1)$$

$$NTU = (-\#P_s)\ln\left\{1 + (1/CR)\ln\left[1 - CR(1 - ER^{1/\#P_s})/(CR - ER^{1/\#P_s})\right]\right\}$$

$$A_{HT} = \frac{NTU \cdot CMIN}{U}$$

For similar designs assume $A_{HT} \propto V_T$

$$U_I \propto U_{II}$$

thus

$$\frac{V_I}{V_{II}} \propto \frac{CMIN_I}{CMIN_{II}} \times \frac{NTU_I}{NTU_{II}}$$

$$V_I \propto \left(\frac{V_{II}}{CMIN_{II}\ NTU_{II}}\right) CMIN_I\ NTU_I$$

The term $\left(\frac{V_{II}}{CMIN_{II}\ NTU_{II}}\right)$ can be evaluated from known data on previous designs to give

$$V_{WHB} = 11.53 * CMIN * NTU$$

But it should be noted that the above equation assumes constant $\left(\frac{\Delta P}{P}\right)_{WHB}$ and boiler power. If as before Vol $\propto A_{HT}$ then from heat transfer

$$A_{HT} \propto \dot{Q} \propto \dot{m}_s \Delta h_s$$

thus

$$V \propto \dot{m}_s \Delta h_s$$

Correcting the above equation for power gives

$$V_{WHB} = .0017\ CMIN * NTU * \dot{m}_s * \Delta h_s$$

As stated previously Vol $\propto \left(\frac{\Delta P}{P}\right)^{-.41}$.

Correcting again for pressure drop yields

$$V_{WHB} = .00045 \text{ CMIN} * \text{NTU} * \dot{m}_s * \Delta h_s \left(\frac{\Delta P}{P}\right)^{-.41} \quad (ft^3)$$

CMIN in BTU/s - ^oR

$\dot{m}_s$ in lbm/s

Δh_s in BTU/lbm

f. Condenser. The condenser is assumed to be a shell and tube crossflow heat exchanger with salt water inside the tubes and steam outside.

The following are defined.

D - inside tube diameter

N - number of tubes

L - length of tubes

g - Gravitational acceleration constant

ΔP_c - pressure drop on the salt water side

f - friction factor

ρ_c - density of cooling water

V_c - velocity of cooling water

A_c - cross sect. free flow area on water side

w_c - flow rate of cooling water

q - heat flow in condenser

C_c - w_c (x) specific heat of cooling water (C_p)

ΔT_c - temp. difference across the cooling water side

X - steam quality entering condenser

h_{fg} - enthapy difference between sat. liquid and sat. steam

$\dot{m}_s$ - mass flow of steam entering condenser

From basic heat transfer relationships

$$A_{HT} = \pi DNL$$

where
$$N = \frac{4 A_c}{\pi D^2}$$

$$L = \frac{g \Delta P_c D}{2 f \rho_c V_c^2}$$

therefore,
$$A_{HT} = \frac{2 A_c g \Delta P_c}{f \rho_c V_c^2}$$

where
$$A_c = \frac{w_c}{\rho_c V_c}$$

$$w_c = q/(C_c \Delta T_c)$$

$$q = \dot{m}_s \lambda h_{fg}$$

thus

$$A_{HT} = \frac{2g\,\Delta P_c\, h_{fg}\, \dot{m}_s X}{f \rho_c^2 V_c^3 C_c \Delta T_c}$$

For similar designs assume $V_T \propto A_{HT}$

thus

$$\frac{V_2}{V_1} = \underbrace{\left(\frac{g\,\Delta P_c\, h_{fg}}{f\rho_c^2 V_c^3 C_c \Delta T_c}\right)_2 \Big/ \left(\frac{g\,\Delta P_c\, h_{fg}}{f\rho_c^2 V_c^3 C_c \Delta T_c}\right)_1}\ \frac{(\dot{m}_s X)_2}{(\dot{m}_s X)_1}$$

≈ 1 for similar designs assuming same ΔP_c

therefore

$$V_2 = \left[\frac{V_1}{(\dot{m}_s X)_1}\right](\dot{m}_s X)_2$$

The quantity in brackets can be evaluated from known data on previous designs (ref. 11) to give

$$V_{cond} = 27.3\, \dot{m}_s X \qquad ft^3$$

$\dot{m}_s$ in lbm/s

IV. DESIGN OF A 25,000 HP PLANT

A proposed design of a 25,000 HP coal-fired combined cycle was completed. Coal combustion is accomplished using a pressurized fluidized bed (PFB). Design parameters were eventually chosen after investigating the effect of changing pressure ratios, varying $\left(\frac{\Delta P}{P}\right)$ in the Brayton cycle, varying E_R (regenerator effectiveness), different compressor inlet temp. (T_1) and steam condensing temp. (T_{15}), and variations in Brayton working pressure (WP) and saturation pressure ($P_{min.}$).

a. The Fluidized Bed. The pressurized fluidized bed operates at a pressure of 3 to 10 atmospheres. Pressure is maintained by an exhaust driven compressor. For proper combustion incoming air must be preheated by the recuperator to at least 700°F. The bed material is primarily limestone which serves to capture sulfer in the coal and thus minimize SO_2 emissions. A bed temperature of 1700°F is generally used to keep NO_x emissions at an acceptable level.

The fluid bed itself is about 4 ft. deep. Preheated combustion air enters the bed via a perforated bed plate underneath the bed. Pressure drop across the bed is assumed to be 10%.

The PFB is used to heat the working fluid in the closed Brayton cycle. The working fluid leaves the regenerator and

enters the tubes in the PFB. The working fluid is first heated by convection from the gas leaving the bed and is then ducted to tubes submerged in the fluid bed for final heating to the turbine inlet temp (T_4). (Figure 7)

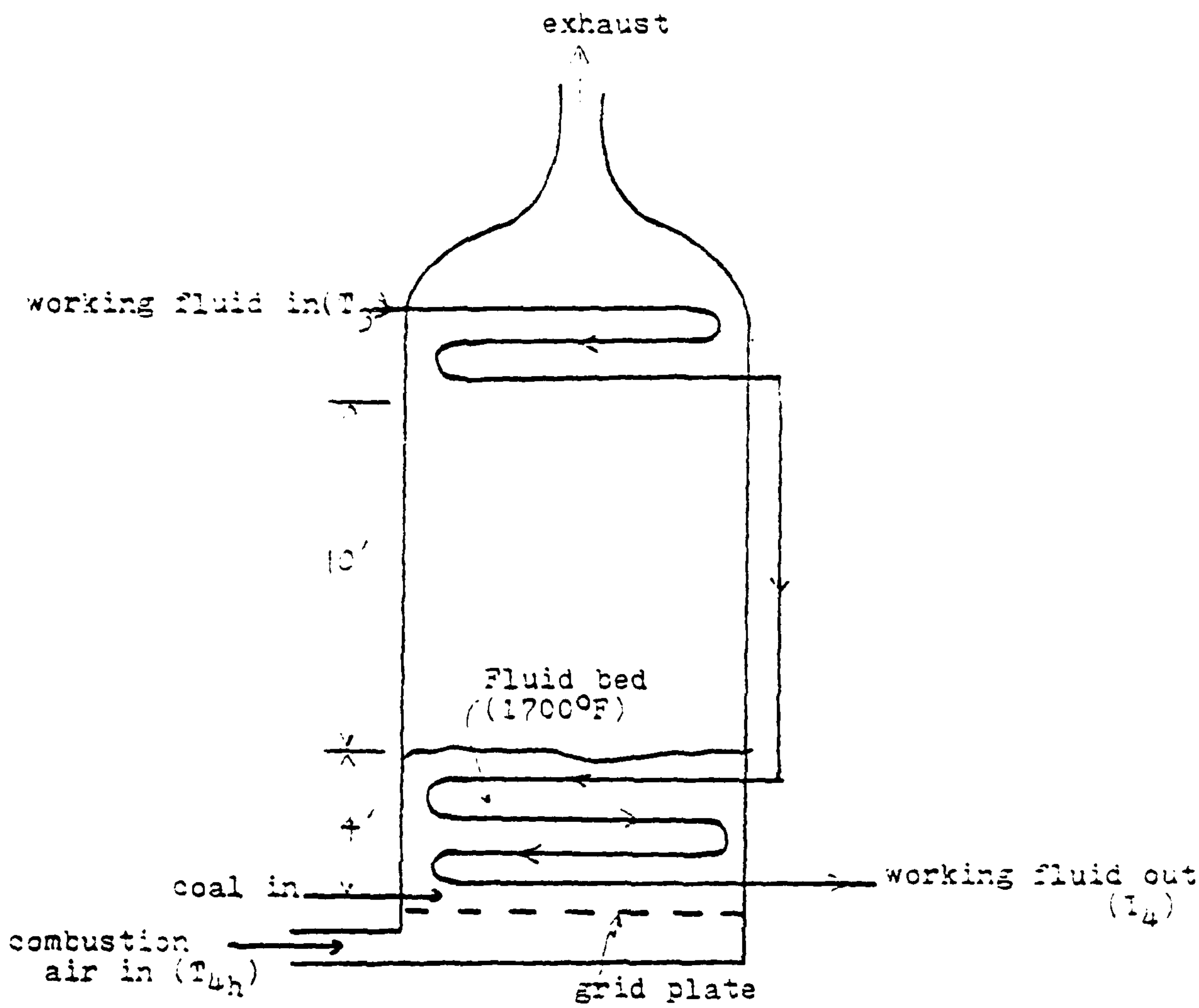


Figure 7.

b. <u>Assumptions</u>. The computer program was run with the following input values:

$\eta_c(P)$	=	.88
$\eta_T(P)$	=	.91
E_R	=	.88
$\eta_p(is)$	=	.80
$\eta_T(is)$	=	.88
$\left(\frac{\Delta P}{P}\right)_T$	=	.10
η_{WHB}	=	.94
T_1	=	540.0 ^oR
T_4	=	2060.0 ^oR
T_{12}	=	540.0 ^oR
ΔT	=	+35.0 ^oR
WP	=	135.0 psi
P_{SAT}	=	200.0 psi (want at least 50 psi greater than WP)
Power	=	25000 HP
T_{1h}	=	535.0 ^oR
T_{7h}	=	760 ^oR (limited by acid corrosion in the turbine)
$\eta_c(P)_h$	=	.85
$\eta_T(P)_h$	=	.89
ϕ	=	.90 (11% excess air)

In addition it was assumed that scoop injection could be used for both the cooler and condenser at speeds greater than 10 knots, therefore power requirements for salt water circulating

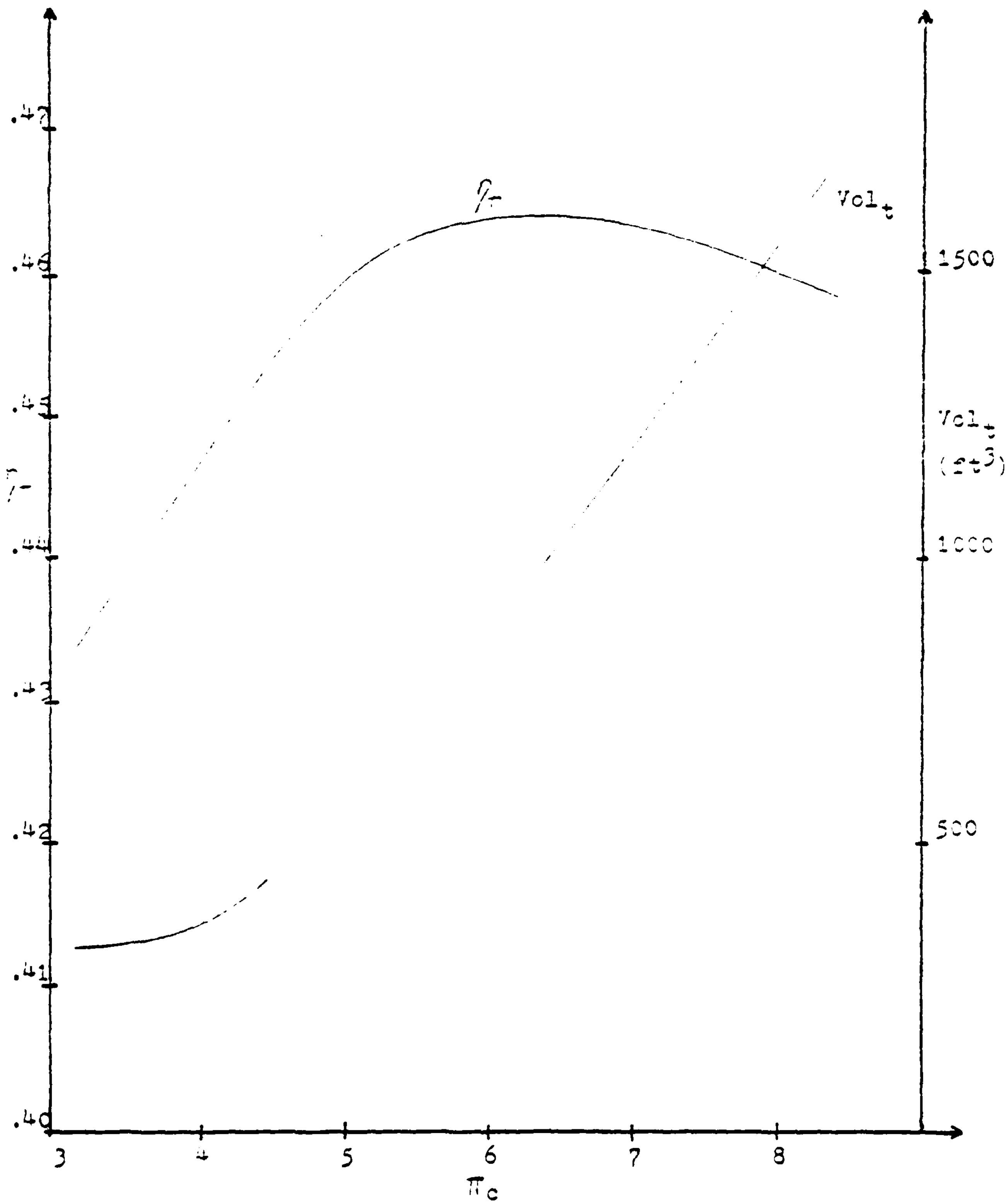
pumps were excluded in the calculations.

c. Results. The computer output is provided in Appendix B. Note that the saturation pressure for pressure ratios of 2 and 3 were below the 200 psia specification. This is due to the fact that the low pressure stream leaving the regenerator is at a temperature below the saturation temp. at 200psia. A plot of η_T vs. π_c and Vol_{total} vs. π_c is shown in Figure 8.

Note that the volume increases sharply above $\pi_c = 4$ due largely to the rapid growth in the waste heat boiler. Maximum η_T occurs at around $\pi_c = 6$. To investigate the effect of $\left(\frac{\Delta P}{P}\right)_T$ on η_T and Vol_t, the program was run for $\left(\frac{\Delta P}{P}\right)_T$ = .04, .06, .08, .10, and .12 between π_c of 4 and 9.

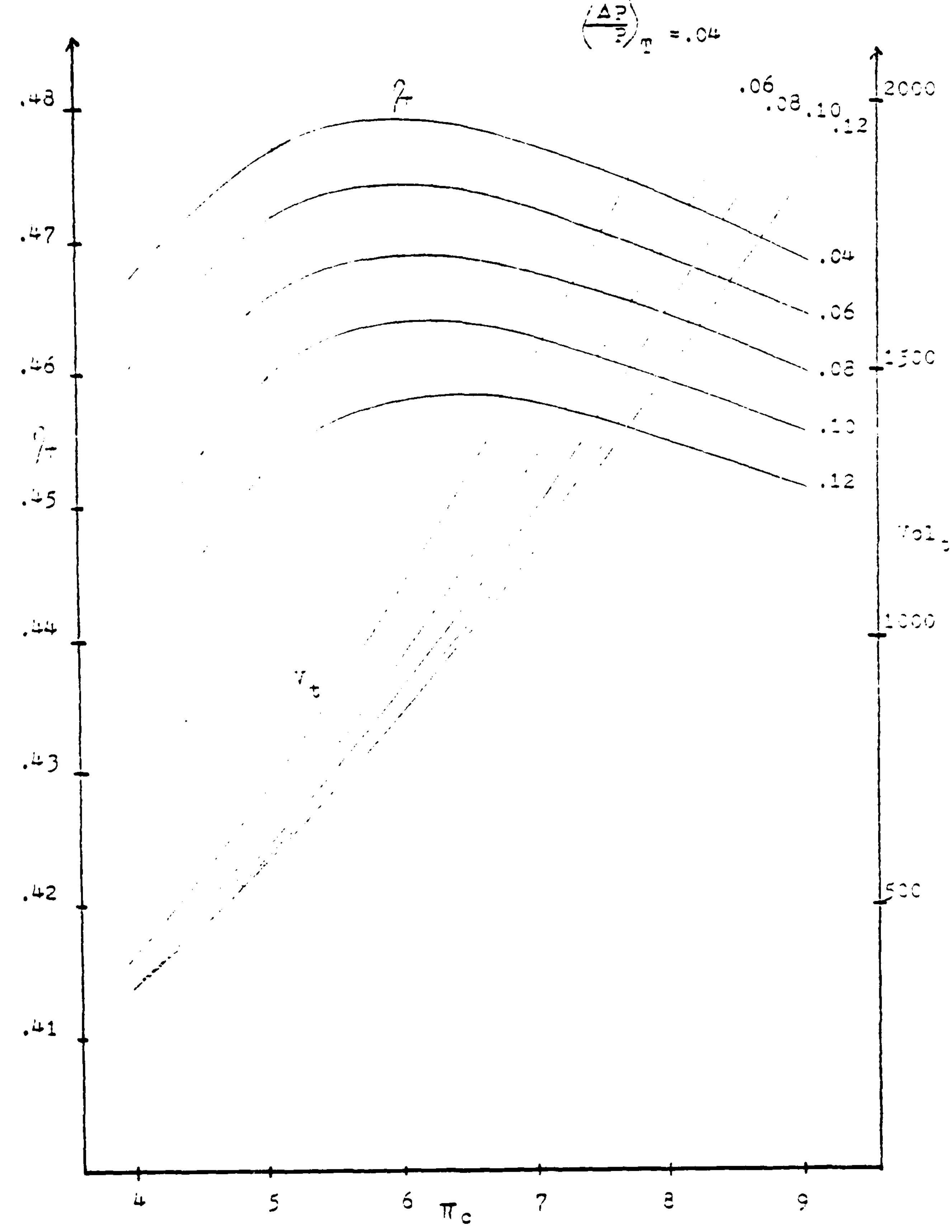
The results plotted on Figure 9 show that significant gains in η_T can be realized with very little increase in V_T for the range of 4-6 pressure ratio. The effect of changing E_R (regenerator effectiveness) in this range with a $\left(\frac{\Delta P}{P}\right)_T$ = .05 was further investigated.

The results listed in Table 1 show no increase in efficiency as E_R is reduced. Furthermore, volume increases sharply as a greater percentage of the work is being done by the Rankine cycle as E_R decreases.
Thus for greatest efficiency with minimum volume, cycle operation should be at the highest E_R practicable.



<u>Figure 3.</u>

Figure 9.



$\left(\frac{\Delta P}{P}\right)_T$ = .04
.06 .08 .10 .12
η
.48
.47
.46
.45
.44
.43
.42
.41
.04
.06
.08
.10
.12
2000
1500
1000
500
Vol_t
V_t
4
5
6
7
8
9
π_c

Table 1

c	E_R	t	Vol_t (ft^3)
3	.88	.452	335
	.80	.423	646
	.50	.402	7038
4	.88	.464	408
	.80	.454	811
	.50	.432	4350
5	.88	.474	724
	.80	.464	1083
	.50	.447	3741
6	.88	.475	1050
	.80	.471	1396
	.50	.456	3508
7	.88	.473	1450
	.80	.471	1750
	.50	.460	3435

To investigate the effect of changes in T_1 (compressor inlet temp.) runs were made on 4 different temperatures and the results shown in Figure 10. As expected efficiency decreased with increasing T_1 with very little change in the range of interest 4-4.5 π_c. Volume on the other hand decreased slightly for π_c less than 3.5 but increased significantly above that. Thus for best efficiency and least volume T_1 should be kept as close to cooling water temperature as possible.

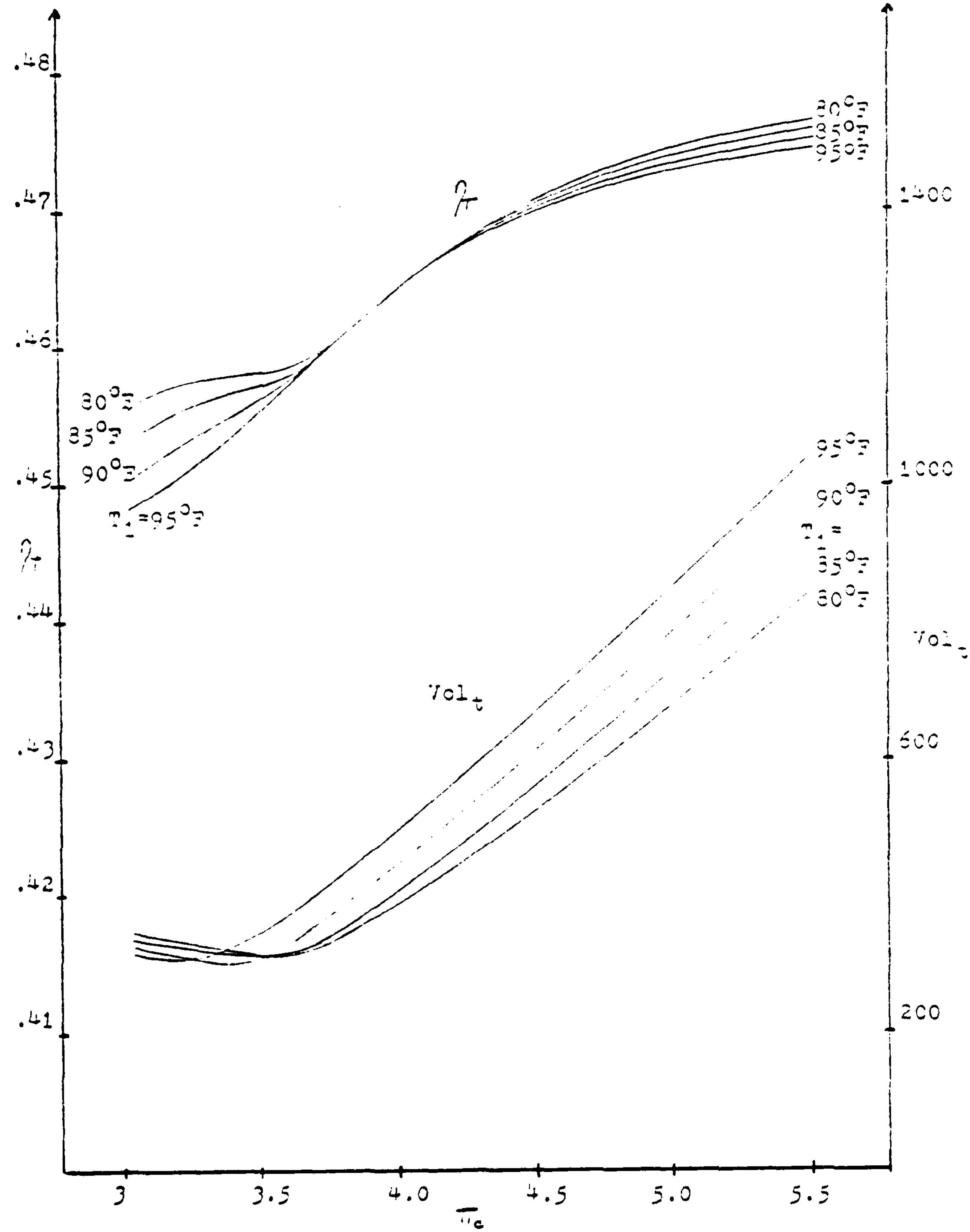
Finally, changes in working pressure of the Brayton cycle were investigated. Decreases in working pressure may require additional volume of Brayton cycle components yet at the same time allow a decrease in Rankine cycle steam pressure (recall $50^{\circ}R$ pressure differential between boiler streams). This will in turn increase cycle efficiency yet further increase volume. These results are born out in Figure 11.

d. <u>System Selection</u>. In accordance with the design criteria the highest thermal efficiency considered is 0.487 (Figure 11). Thus it is desired to select the plant with minimum volume at $\eta_t = .482$.

Figure 11 shows that only those alternatives with Brayton cycle working pressures of 70psi and 100psi meet the efficiency requirement. Of these two the 100psi case has minimum volume at desired efficiency.

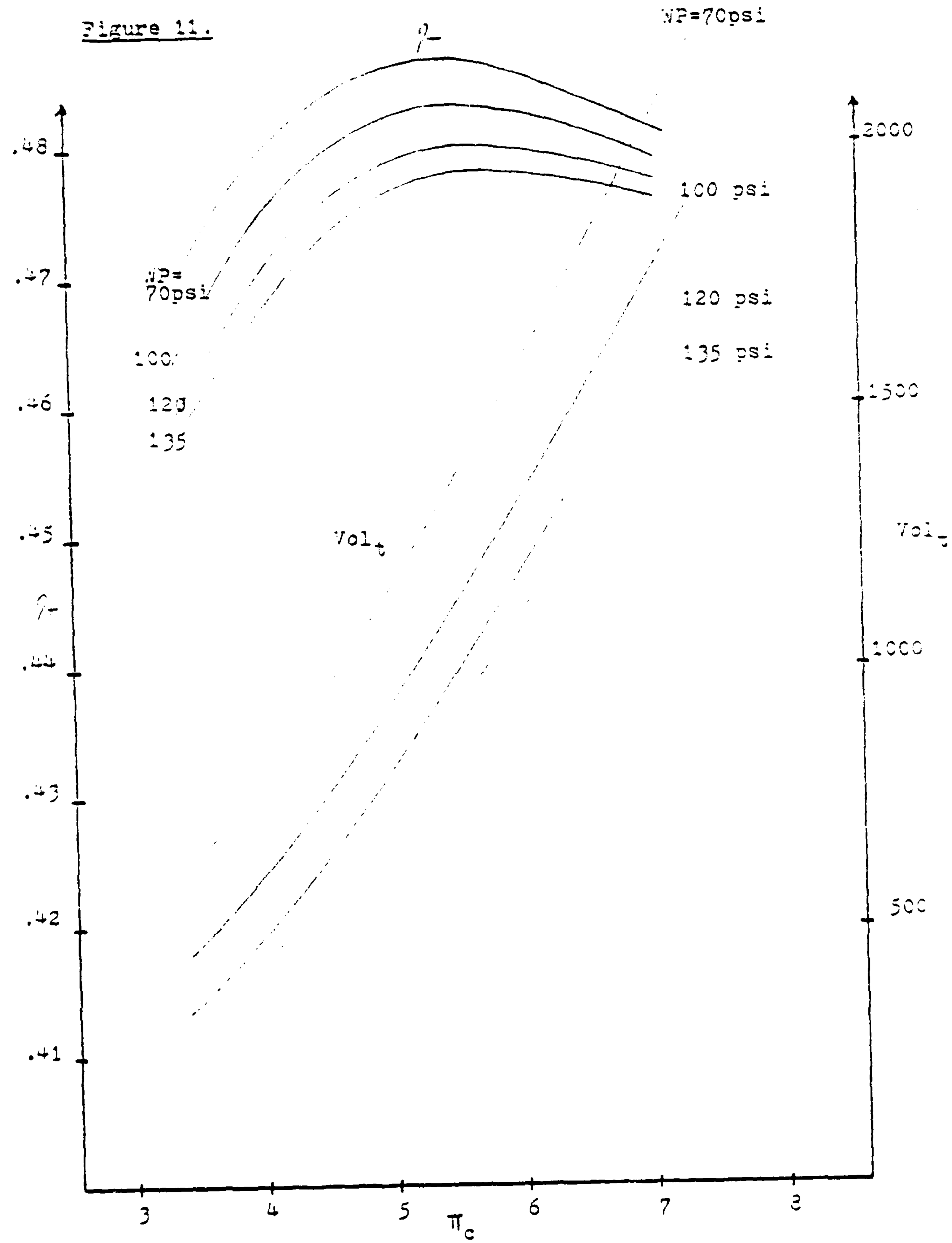
However, the problem does not end here. If one looks at

Figure 10.



.48
.47
.46
.45
.44
.43
.42
.41
η_t
80°F
85°F
90°F
T_1=95°F
η_t
80°F
85°F
95°F
1400
1000
600
200
95°F
90°F
T_1 =
85°F
80°F
Vol_t
Vol_t
3
3.5
4.0
4.5
5.0
5.5
π_c

Figure 11.



η
WP=70psi
100 psi
120 psi
135 psi
WP=
70psi
100
120
135
.48
.47
.46
.45
.44
.43
.42
.41
η
Vol$_t$
Vol$_t$
2000
1500
1000
500
3
4
5
6
7
8
π_c

the steam quality leaving the turbine at the π_c = 5, WP = 100 psi point it is of the order of 0.84. This is a little lower than normally desired for design. In fact, the steam quality of all alternatives thus far considered within the range of interest is low. If a steam quality of order .87 is desired the current design pt. can change in one of 3 ways as follows:

1) Raise steam condensing temp (T_{1S})

2) Lower minimum saturation pressure (P_{min})

3) or a combination of the above two

Through trial and error it was found that case 1 gave the greatest benefit with the least penalty. Computer runs were made in the 4-5 π_c range with the steam condensing temp raised to 560°R for both the 70 and 100psi alternatives. The results show a substantial increase in quality, slight decrease in efficiency and very little change in total volume (Table 2).

(psi) Working Pressure	π_c	η_t	x	Vol_t (ft^3)
70	4.5	.480	.864	948.3
70	5.0	.482	.871	1140.6
100	5.0	.479	.862	956.5

Table 2.

The 70 psi designs have better steam quality and η_t than the best 100 psi case. In addition, at 4.5 π_c the volume is lower. Between the two 70 psi cases above a trade-off exists between η_t and Vol_t. Assume that volume comparable to the case

of $\pi_c = 5$, WP = 100 psi, $T_{1S} = 540^oR$ discussed above (Figure 11) is desired. If this is the case then the new design point is at a Brayton cycle working pressure of 70 psi, $\pi_c = 4.5$, $T_{1S} = 560^oR$. The results of this design are summarized in the following computer printout:

BRAYTON CYCLE: PRESSURE RATIO = 4.50
WORKING PRESSURE = 70.0
TOTAL PRESSURE DROP = .05
MASS FLOW (LBM/S)=206.93

SUMMARY: THERMAL EFF=0.480
HEATER EFF=0.930 PHI=0.901
SFC(LBM/HP-HR)=0.440
VOL TOTAL(CU FT)= 948.3
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.101

STATE	TEMP(DEG R)
1	540.0
2	876.0
3	1403.9
4	2060.0
5	1475.9
6	948.0
PP	836.2
7	809.2

COMPONENT	DPP	DHD	VOL(CU FT)	
REGEN	.010	.950	26.5	EFF=0.88
TURBINE			5.5	EFF(P=0.91
COOLER	.009	.490	193.1	
COMPRESSOR			29.7	EFF(P)=0.88
HEATER	.0006			
BOILER	.030			EFF=0.94

HEATER CYCLE: PRESSURE RATIO=4.79
TOTAL PRESSURE DROP =.43
AIR FLOW(LBM/S)= 36.85
FUEL FLOW(LBM/S)= 3.06

STATE	TEMP(DEG R)
1H	535.0
2H	835.2
3H	1440.5
4H	4809.8
5H	1503.9
6H	945.0
7H	760.0

COMPONENT	DPP	DHD	VOL(CU FT)	
RECUP	.313	.650	1.9	EFF=0.91
TURBINE			0.7	EFF(P)=0.89
HEATER	.019	.430	31.4	
COMPRESSOR			10.8	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE = 120.0
STEAM FLOW(LBM/S)= 5.69
FLOW RATIO =0.0275

STATE	TEMP(DEG R)	PRESSURE(PSI)	
1S	560.0	0.9	
2S	575.0	120.0	
SF	801.2	120.0	
3S	913.0	120.0	
4S	560.0	0.9	QUAL=0.864

COMPONENT	VOL(CU FT)	
BOILER	507.9	
TURBINE	6.5	EFF(IS)=0.88
CONDENSER	134.2	

V. APPLICATION

The application of this system to Navy ships has many implications. One of which is the fact that neither Closed Brayton nor combined cycles have been previously used. However, there is a considerable effort at this time to get COGAS (Combined Gas Turbine and Steam) systems on board Navy ships. These systems are all limited to burning oil fuels. The Closed Brayton combined cycle when equipped with a multi-fuels heater offers the additional flexibility of burning the cheapest or most readily available fuel.

Advantages of the Closed Brayton combined cycle are:

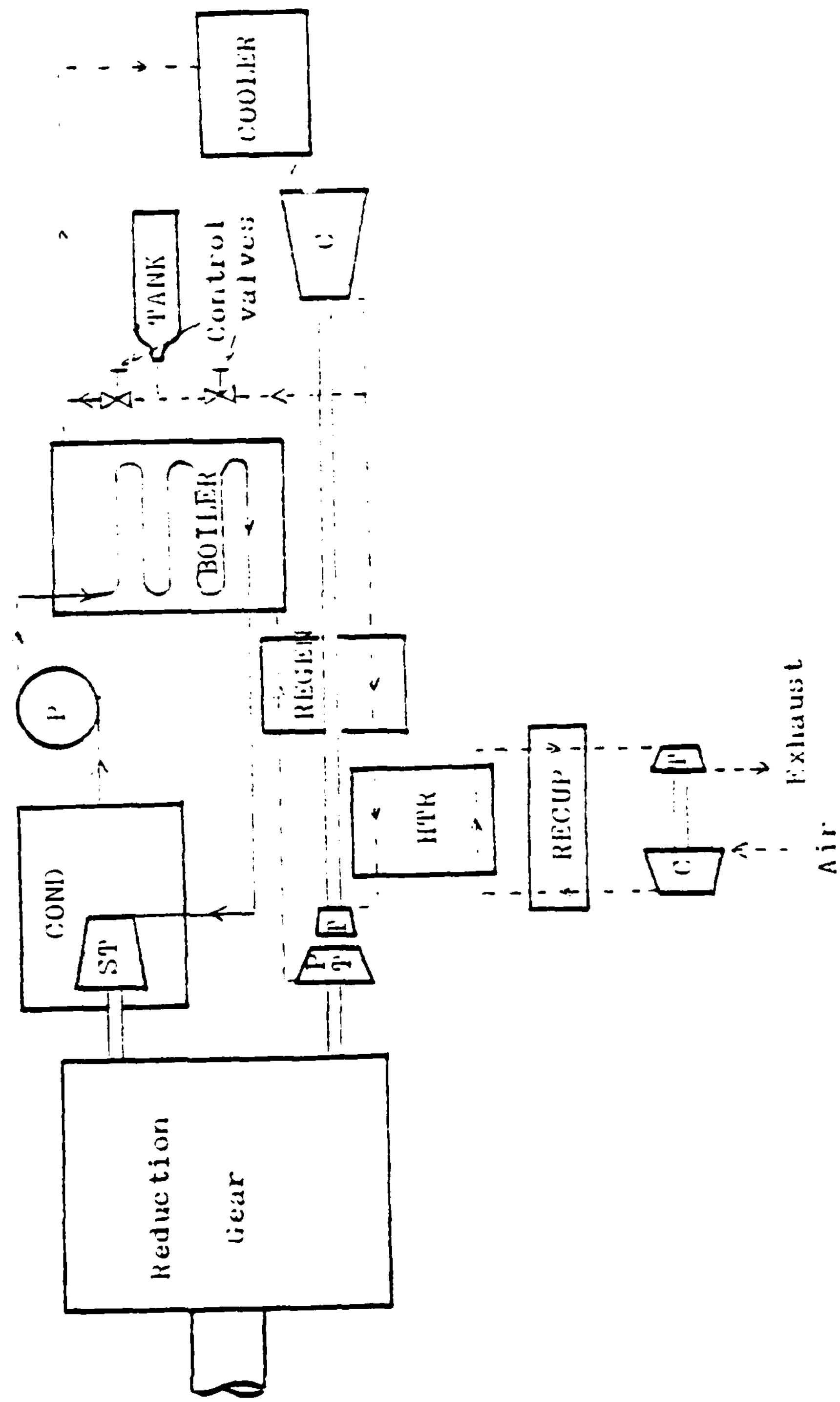
1) The Brayton cycle operates at temperatures substantially below the material limit than in the open cycle gas turbines resulting in cheaper, more reliable, and longer life components.

2) Low pressure steam systems are well proven, and have significantly less water chemistry problems than higher pressure ones.

3) As load decreases and fuel flow is cut back, the control valves (Figure 12) between the compressor outlet and cooler inlet can be opened as necessary (keeping max. temp and speed constant) so as to ensure that the vector diagrams of all sections of the turbomachinery blading remain unchanged. Turbine and compressor efficiencies will remain essentially the same producing high η_t at part load. This requires use of a CRP

Figure 12.

Functional Component Arrangement



(controllable reversible pitch) propeller to maintain constant shaft speed at part loads.

Disadvantages of the Closed Brayton combined cycle are:

1) Higher overall system weight and volume due to the additional heat exchangers and piping required for the Closed Brayton cycle.

2) A complex, integrated control system is required to coordinate control of the combined cycle, fuel flow, and CRP at part loads.

3) Fluidized bed combustion is not a fully developed engineering concept. Land based prototypes have been built and operated with limited but increasing levels of success. A shipboard prototype is perhaps 5 years in the future with production of a marinized version at least 10 years away.

4) If coal is to be used as the primary fuel, it should be noted that it has a lower heating value per ton than oil. The result is a larger potentially more expensive (greater acquisition cost) ship for the same military mission.

5) Reaction time with coal fired fluidized bed combustors to load changes are generally slower than that for oil heaters necessitating in a separate fuel oil system for rapid increases in load.

VI. CONCLUSIONS AND RECOMMENDATIONS

The Closed Brayton combined cycle offers an attractive alternative system for many shipboard applications. As with the first Arab oil embargo, subsequent cut-offs of Arab oil could result in decreased steaming days and a lower overall readiness of the U.S. Navy's defense posture.

If our ships could have the option of burning the most readily available fuel, this reduction in readiness can be avoided by switching to coal, a fuel source in great abundance in the U.S.

Paramount in the engineering of a successful multi-fuels propulsion plant is the development of a marinized fluid bed combustor. The design of heat exchangers and other turbo-machinery used in the combined cycle are well within the state of the art of present technology.

It is thus that the author concludes the following:

1) The design of a modern coal burning combined cycle is within the reach of present technology.

2) This combined cycle would offer high efficiency at both rated load and part load due to the unique controlling aspect of the Closed Brayton cycle.

3) Weight and volume of a Closed Brayton combined cycle would be greater than that of an equivalently rated COGAS type system.

4) Coal offers the benefit of a readily available,

relatively cheap fuel at the expense of increased ship size and acquisition cost due to lower BTU heat value per ton when compared to oil.

The author makes the following recommendations:

1) Development of a land based test site of a Closed Brayton combined cycle equipped with a fluid bed heater should be considered.

2) A pressurized fluidized bed combustor suitable to marine applications should be developed.

3) A shipboard prototype system should be integrated on a U.S. Navy ship to attempt to quantitatively define the shipboard impact of the system.

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APPENDIX A

The computer program performs a thermodynamic analysis and component volume computation for various specified Brayton cycle compressor pressure ratios.

The program accepts as input the following quantities:

$\eta_c(P)$	-	polytropic compressor efficiency (Brayton cycle)
$\eta_T(P)$	-	polytropic turbine efficiency (Brayton cycle)
E_R	-	regenerator effectiveness
$\eta_{P(is)}$	-	isentropic pump efficiency (Rankine cycle)
$\left(\frac{\Delta P}{P}\right)_T$	-	total pressure drop around Brayton cycle loop
η_{WHB}	-	efficiency of the waste heat boiler
T_1	-	compressor inlet temp. (Brayton cycle)
T_4	-	turbine inlet temp (Brayton cycle)
T_{1S}	-	steam condensing temp.
ΔT	-	min. temp diff. between streams in WHB (at PP and T_6)
P_{SAT}	-	min. saturation pressure in the Rankine cycle
Power	-	net power out of the combined cycle
T_{1h}	-	compressor inlet temp. (heater cycle)
T_{7h}	-	turbine exit temp. (heater cycle)
$\eta_c(P)_h$	-	polytropic compressor efficiency (heater cycle)
$\eta_T(is)$	-	isentropic turbine efficiency (Rankine cycle)

$\eta_{(R)t}$ - polytropic turbine efficiency (heater cycle)

ϕ - equivalence ratio

WP - working pressure in the Brayton cycle.

Output is as shown in Appendix B.

The main program performs the thermodynamic analysis of the closed Brayton cycle and the Rankine cycle. Functions performed by the various subfunctions are as follows:

1) Subroutine HEATER performs a thermodynamic analysis of the heater cycle.

2) Subroutine DHD computes $\frac{D_h}{D}$ for the regenerator, recuperator, cooler and heater.

3) Subroutine DPP calculates $\frac{\Delta P}{P}$ for the regenerator, recuperator, cooler, heater and boiler.

4) Subroutine TURVOL finds turbine volumes.

5) Subroutine COMPVOL finds compressor volumes.

6) Subroutine REGEN computes the regenerator volume.

7) Subroutine RECUP computes the recuperator volume.

8) Subroutine COOLER computes the cooler volume.

9) Subroutine COMBUSTOR computes the heater volume.

10) Subroutine WHBVOL computes the waste heat boiler volume.

11) Subroutine CONDVOL computes the condenser volume.

12) Subroutine STMTAB is a computerized version of the steam tables. Reproduced from ref. 15.

```
      DIMENSION CR(30),SATLIQ(3),STEAM(3),TPP(12),WR(11)
      REAL NCP,NTP,NPIS,NTIS,NWHB,NT,L1,L2,KO
      REAL NCPH,NTPH,MB,MS,MA,MF,NH,N,IT,KI1,NPP
      REAL LHV,M,M1,M2,NTU
      COMMON /STM/PRESS,TEMP,QUAL,SATLIQ,STEAM
      COMMON /HTR/NH,PRH,DPPH,T2H,T3H,T4H,T5H,T6H,MA,MF
      COMMON /DIA/DHDREG,DHDCLR,DHDHTR,DHDREC
      COMMON /PRES/DPPWHB,DPPREG,DPPCLR,DPPREC,DPH1,DPH2,DHDHT
      COMMON /COMB/DPPRG,DPPRC,DPPHE,DHDHR,DPHE2
      READ(5,10) NCP,NTP,ER,NPIS,NTIS,DP,NWHB,T1,T4,T1S,DT,PMIN,K
10    FORMAT (7F4.3,5F7.1,I2)
      READ(5,15) PWR,T1H,T7H,NCPH,NTPH,PHI,WP
15    FORMAT (F8.1,2F6.1,3F5.3,F6.1)
      READ(5,20) (CR(I),I=1,K)
20    FORMAT (<K>F6.2)
      R=.068544
      DO 200 I=1,K
        PR=CR(I)
        Z=CP1(T1)+(R/NCP)*LOG(PR)
        T2=EXP(Z/.2475)
        CALL ITER (T2,Z)
        TR=1/(1-DP)/PR
        Z=CP1(T4)+R*NTP*LOG(TR)
        T5=EXP(Z/.2475)
        CALL ITER (T5,Z)
        IF(T2 .GT. T5) GO TO 30
        T6=T5-ER*(T5-T2)
        T3=T2+ER*(T5-T2)
        GO TO 40
30      T6=T5
        T3=T2
40      L1=T1
        L2=1122.0
        IF(L2 .GT. T6) GO TO 50
        GO TO 60
50      L2=T6-(T6-T1)/10.
60      TPP(1)=L2
        WR1=0.
        C=(L2-L1)/10.
        DO 70 J=1,11
          TSF=TPP(J)-DT
          T3S=T6-DT
          T=TSF-460
          CALL STMTAB (2,T,0.)
          PSF=PRESS
          HSF=SATLIQ(2)
          T=T3S-460
          CALL STMTAB (7,PSF,T)
          H3S=STEAM(2)
          S3S=STEAM(3)
          T=T1S-460
```

```
      CALL STMTAB (2,T,0.)
      P4S=PRESS
      V1S=SATLIQ(1)
      H1S=SATLIQ(2)
      S1S=SATLIQ(3)
      CALL STMTAB (9,P4S,S3S)
      H4SIS=STEAM(2)
      H4S=H3S-NTIS*(H3S-H4SIS)
      CALL STMTAB (8,P4S,H4S)
      X=QUAL
      V4S=STEAM(1)
      H2SIS=H1S+V1S*(PSF-P4S)
      H2S=H1S+(H2SIS-H1S)/NPIS
      T2S=T1S+15.
      B=NWHB*(CP2(T6)-CP2(TPP(J)))/(H3S-HSF)
      Z=ACP(TPP(J))*TPP(J)-B*(HSF-H2S)/NWHB
      T7=TPP(J)
      CALL ITER2 (T7,Z)
      WR(J)=B*(H3S+H1S-H4S-H2S)
      IF(J .EQ. 1) GO TO 68
      IF(T7 .LT. T1) GO TO 75
      IF(PSF .LT. PMIN) GO TO 75
      IF(WR(J) .GT. WR1) GO TO 68
      IF(X .GT. .87) GO TO 75
68    WR1=WR(J)
      TPP(J+1)=TPP(J)-C
70    CONTINUE
      GO TO 76
75    IF(ABS(TPP(J)-TPP(J-1)) .LT. .1) GO TO 76
      L1=TPP(J)
      L2=TPP(J-1)
      GO TO 60
76    WB=CP2(T4)+CP2(T1)-CP2(T5)-CP2(T2)
      QIN=CP2(T4)-CP2(T3)
      NT=(WB+WR(J))/QIN
      MB=.707*PWR/(WB+WR(J))
      MS=B*MB
      QB=MB*QIN
      CALL HEATER (QB,T1H,T7H,NCPH,NTPH,PHI,T3,T4)
      CALL DHD(PR)
      DHDHT=DHDHTR
      K=1
77    CALL DPP(T6,T7,PR,DP,ER,T2,T3,DPPH,T4H,T5H,T4,PRH,WP,MB,MA
     * ,MF,K)
      G=1.33
      N=G/(G-(G-1.)*NTP)
      P=WP*(PR-DPPREG)
      V=1./ARHO(T4,P)
      VMNTURB=0.0
      CALL TURBVOL (N,MB,V,TR,T4,VMNTURB)
      G=1.39
```

```
      N=G/(G-(G-1.)*NTPH)
      P=14.7*(PRH-DPPREC)
      V=1./ARHO(T6H,P)
      M=MA+MF
      VHTRTURB=0.0
      TRH=1./(1.-DPPH)/PRH
      CALL TURBVOL (N,M,V,TRH,T6H,VHTRTURB)
      T=T1S/T3S
      P=P4S/P5F
      N=1./(1.-LOG(T)/LOG(P))
      VSTMTURB=0.0
      CALL TURBVOL (N,MS,V4S,P,T3S,VSTMTURB)
      G=1.4
      N=G/(G-(G-1.)/NCP)
      V=1./ARHO(T1,WP)
      VMNCOMP=0.0
      CALL COMPVOL (N,MB,V,PR,VMNCOMP)
      V=1./ARHO(535.,14.7)
      VHTRCOMP=0.0
      CALL COMPVOL (N,MA,V,PRH,VHTRCOMP)
      D=.125
      VREGEN=0.0
      CALL REGEN (T2,T3,T5,T6,WP,D,MB,PR,DPPREG,DHDREG,VREGEN)
      VRECUP=0.0
      CALL RECUP (T2H,T3H,T5H,T6H,PRH,DPPH,D,MA,MF,DPPREC,VRECUP)
      D=.25
      VCLR=0.0
      CALL COOLER (T7,T1,WP,D,NTP,MB,DHDCLR,DPPCLR,VCLR)
      DPPRG=DPPREG
      DPPRC=DPPREC
      DPPHE=DPPH
      DHDHR=DHDHTR
      DPHE2=DPH2
      VHTR=0.0
      CALL COMBUSTOR (T3,T4,T4H,T5H,WP,PR,D,MB,PRH,MA,MF,VHTR)
      VWHB=0.0
      CALL WHBVOL (MB,MS,T6,T7,NWHB,H2S,H3S,DPPWHB,VWHB)
      VCOND=0.0
      CALL CONDVOL (MS,X,VCOND)
      VOLT=VMNTURB+VHTRTURB+VSTMTURB+VMNCOMP+VHTRCOMP+VREGEN+VRECUP
     * +VCLR+VHTR+VWHB+VCOND
      IF(K .EQ. 1) GO TO 78
      IF(VOLT .GT. VOL1) GO TO 81
      IF((DPPCLR-.002) .LT. 0.0) GO TO 81
78    K=K+1
      VOL1=VOLT
      V1=VCLR
      V2=VWHB
      GO TO 77
81    VOLT=VOL1
      VCLR=V1
```

```
      VWHB=V2
      WRITE(6,82) PR,WP,DP,MB
82    FORMAT('1','BRAYTON CYCLE: PRESSURE RATIO=',F5.2/1X,'WORKING
     * PRESSURE=',F5.1/1X,'TOTAL PRESSURE DROP=',F3.2/1X,'MASS FLOW
     * (lbm/s)=',F6.2//10X,'STATE  TEMP(deg R)')
      WRITE(6,83) T1
83    FORMAT(12X,'1',5X,F6.1)
      WRITE(6,84) T2
84    FORMAT(12X,'2',5X,F6.1)
      WRITE(6,85) T3
85    FORMAT(12X,'3',5X,F6.1)
      WRITE(6,86) T4
86    FORMAT(12X,'4',5X,F6.1)
      WRITE(6,87) T5
87    FORMAT(12X,'5',5X,F6.1)
      WRITE(6,88) T6
88    FORMAT(12X,'6',5X,F6.1)
      WRITE(6,89) TPP(J)
89    FORMAT(12X,'PP',5X,F6.1)
      WRITE(6,90) T7
90    FORMAT(12X,'7',5X,F6.1//5X,'COMPONENT  DPP  DHD  VOL(cu ft)')
      WRITE(6,91) DPPREG,DHDREG,VREGEN,ER
91    FORMAT(7X,'REGEN',4X,F4.3,1X,F4.3,1X,F7.1,2X,'EFF=',F4.2)
      WRITE(6,92) VMNTURB,NTP
92    FORMAT(6X,'TURBINE',15X,F5.1,2X,'EFF(P)='F4.2)
      WRITE(6,93) DPPCLR,DHDCLR,VCLR
93    FORMAT(6X,'COOLER',4X,F4.3,1X,F4.3,3X,F5.1)
      WRITE(6,94) VMNCOMP,NCP
94    FORMAT(4X,'COMPRESSOR',14X,F5.1,2X,'EFF(P)=',F4.2)
      WRITE(6,95) DPH1
95    FORMAT(6X,'HEATER',4X,F5.4)
      WRITE(6,96) DPPWHB,NWHB
96    FORMAT(6X,'BOILER',4X,F4.3,15X,'EFF=',F4.2/)
      WRITE(6,100) PRH,DPPH,MA,MF
100   FORMAT(1X,'HEATER CYCLE: PRESSURE RATIO=',F5.2/1X,'TOTAL
     * PRESSURE DROP=',F3.2/1X,'AIR FLOW(lbm/s)=',F6.2/1X,'FUEL
     * FLOW(lbm/s)=',F6.2//10X,'STATE  TEMP(deg R)')
      WRITE(6,101) T1H
101   FORMAT(12X,'1h',5X,F6.1)
      WRITE(6,102) T2H
102   FORMAT(12X,'2h',5X,F6.1)
      WRITE(6,103) T3H
103   FORMAT(12X,'3h',5X,F6.1)
      WRITE(6,104) T4H
104   FORMAT(12X,'4h',5X,F6.1)
      WRITE(6,105) T5H
105   FORMAT(12X,'5h',5X,F6.1)
      WRITE(6,106) T6H
106   FORMAT(12X,'6h',5X,F6.1)
      WRITE(6,107) T7H
107   FORMAT(12X,'7h',5X,F6.1//5X,'COMPONENT  DPP  DHD  VOL(cu ft)'
```

```
     * )
      ERH=(T3H-T2H)/(T5H-T2H)
      WRITE(6,110) DPPREC,DHOREC,VRECUP,ERH
110   FORMAT(7X,'RECUP',4X,F4.3,1X,F4.3,1X,F7.1,2X,'EFF=',F4.2)
      WRITE(6,111) VHTRTURB,NTPH
111   FORMAT(6X,'TURBINE',15X,F5.1,2X,'EFF(P)=',F4.2)
      WRITE(6,112) DPH2,DHOHTR,VHTR
112   FORMAT(6X,'HEATER',4X,F4.3,1X,F4.3,3X,F5.1)
      WRITE(6,113) VHTRCOMP,NCPH
113   FORMAT(4X,'COMPRESSOR',14X,F5.1,2X,'EFF(P)=',F4.2/)
      WRITE(6,120) PSF,MS,B
120   FORMAT(1X,'RANKINE CYCLE: SATURATION PRESSURE=',F6.1/1X,
     * 'STEAM FLOW(lbm/s)=',F6.2/1X,'FLOW RATIO=',F6.4//10X,'STATE
     * TEMP(deg R)  PRESSURE(psi)')
      WRITE(6,121) T1S,P4S
121   FORMAT(12X,'1s',5X,F6.1,8X,F6.1)
      WRITE(6,122) T2S,PSF
122   FORMAT(12X,'2s',5X,F6.1,8X,F6.1)
      WRITE(6,123) TSF,PSF
123   FORMAT(12X,'SF',5X,F6.1,8X,F6.1)
      WRITE(6,124) T3S,PSF
124   FORMAT(12X,'3s',5X,F6.1,8X,F6.1)
      WRITE(6,125) T1S,P4S,X
125   FORMAT(12X,'4s',5X,F6.1,8X,F6.1,2X,'QUAL=',F5.3/)
      WRITE(6,128) VWHB
128   FORMAT(5X,'COMPONENT  VOL(cu ft)'/6X,'BOILER',5X,F6.1)
      WRITE(6,129) VSTMTURB,NTIS
129   FORMAT(6X,'TURBINE',5X,F5.1,2X,'EFF(IS)=',F4.2)
      WRITE(6,130) VCOND
130   FORMAT(5X,'CONDENSER',4X,F5.1/)
      WRITE(6,135) NT
135   FORMAT('1','SUMMARY:THERMAL EFF=',F5.3)
      WRITE(6,136) NH,PHI
136   FORMAT(9X,'HEATER EFF=',F5.3,'  (PHI=',F4.2,')')
      SFC=MF*3600./PWR
      WRITE(6,137) SFC,VOLT,PWR
137   FORMAT(9X,'SFC(lbm/HP-hr)=',F5.3/9X,'VOL TOTAL(cu ft)=',F6.1/
     *9X,'POWER(HP)=',F7.1)
      RANKINE=WR(J)/WB
      WRITE(6,138) RANKINE
138   FORMAT(9X,'WORK FRAC BY STM CYCLE=',F5.3)
200    CONTINUE
      STOP
      END
      FUNCTION CP1(T)
      CP1=.2475*LOG(T)-(3.759E-5*T)+(2.553E-8*T**2)-(4.103E-12*T**3)
      RETURN
      END
      FUNCTION CP2(T)
      CP2=.2475*T-(1.379E-5*T**2)+(1.702E-8*T**3)-(3.078E-12*T**4)
      RETURN
```

```
      END
      FUNCTION AMU(T)
      TK=5.*T/9.
      AMU=TK**1.5*1.E-6/(TK+110.4)
      RETURN
      END
      FUNCTION ARHO(T,P)
      R=53.35
      ARHO=P*144./R/T
      RETURN
      END
      FUNCTION AK(T,P)
      A=.6325E-5
      B=245.4
      C=12.
      TK=5.*T/9.
      AKO=(A*TK**.5)/(1.+B*10.**(-1.*C/TK)/TK)*418.6
      AK=(AKO+(1.464E-5*(16.02*ARHO(T,P))**1.23))*.578/3600.
      RETURN
      END
      FUNCTION APR(T,P)
      CP=ACP(T)
      XMU=AMU(T)
      XK=AK(T,P)
      APR=CP*XMU/XK
      RETURN
      END
      FUNCTION ASIGMA(T,P)
      ASIGMA=17.214*(6223.*AK(T,P))**(-1.41)*(1.46*AMU(T))**1.23*
     *(1./APR(T,P)**.564)*(1./(16.02*ARHO(T,P))**.41)
      RETURN
      END
      FUNCTION ACP(T)
      ACP=.2475-3.759E-5*T+5.106E-8*T**2-1.231E-11*T**3
      RETURN
      END

      SUBROUTINE ITER (T,X)
 10   T1=T
      T=EXP((X+.2475*LOG(T)-CP1(T))/.2475)
      IF(ABS(T-T1) .LT. 1.E-2) GO TO 20
      GO TO 10
 20   RETURN
      END
      SUBROUTINE ITER2(T,X)
 10   T1=T
      T=X/ACP(T)
      IF(ABS(T-T1) .LT. 1.E-2) GO TO 20
      GO TO 10
 20   RETURN
      END
```

```
      SUBROUTINE HEATER (QB,T1H,T7H,NCPH,NTPH,PHI,T3,T4)
      REAL LHV,NH,MF,MA,NCPH,NTPH,N
      COMMON /HTR/NH,PRH,DPPH,T2H,T3H,T4H,T5H,T6H,MA,MF
      LHV=12900.
      FA=.0922
      H7H=ACP(T7H)*T7H
      H1H=ACP(T1H)*T1H
      NH=1-((1+FA*PHI)*H7H-H1H)/(FA*PHI*LHV)
      MF=QB/(NH*LHV)
      MA=MF/(FA*PHI)
      T2H=0.0
      T6H=0.0
10    CPT=ACP((T6H+T7H)/2.)
      CPC=ACP((T1H+T2H)/2.)
      N=(1+FA*PHI)*CPT*NTPH*(CPT*T7H-CPC*T1H)
      D=CPC*T1H*(CPC/NCPH-CPT*NTPH)
      PRH=(N/D)**(NCPH/CPC)
      TH2=T1H*PRH**(CPC/NCPH)
      N=(1+FA*PHI)*CPT*T7H+CPC*T1H*(PRH**(CPC/NCPH)-1)
      D=(1+FA*PHI)*CPT*T7H*PRH**(CPT/NTPH)
      DPPH=1-(N/D)**(1./(CPT*NTPH))
      TRH=(1-DPPH)*PRH
      TH6=T7H*(TRH)**(CPT*NTPH)
      IF(ABS(T2H-TH2) .LT. .1) GO TO 20
      IF(ABS(T6H-TH6) .LT. .1) GO TO 20
      T2H=TH2
      T6H=TH6
      GO TO 10
20    T2H=TH2
      T6H=TH6
      T5H=T3+100.
      T4H=T5H+QB/(ACP(T5H/2+1300.)*(MA+MF))
      T3H=T2H+(1+MF/MA)*(T5H-T6H)
      RETURN
      END
      SUBROUTINE DHD(PR)
      COMMON /DIA/DHDREG,DHDCLR,DHDHTR,DHDREC
      DHDREG=.557+.0933*PR
      IF(DHDREG .GT. .95) DHDREG=.95
      DHDCLR=.076+.092*PR
      DHDHTR=.214+.048*PR
      DHDREC=.65
      RETURN
      END

      SUBROUTINE DPP(T6,T7,PR,DP,ER,T2,T3,DPPH,TH4,TH5,T4,PRH,P,M,MA
     *,MF,K)
      REAL M,MA,MF,M1,M2
      COMMON /PRES/DPPWHB,DPPREG,DPPCLR,DPPREC,DPH1,DPH2,DHDHT
```

```
DTG=T6-T7
DPPWHB=FLOAT(K)*.002
DPPREG=(.135-.0284*PR)*DP/.1
IF(DPPREG .LT. .01) DPPREG=.01
IF(T2 .EQ. T3) DPPREG=0.0
DPH2=(.0108*PR-.01)*DP/.1
T1=(T3+T4)*.5
TT=(TH4+TH5)*.5
P1=P*(PR-DP)
P2=14.7*(PRH-DPPH)
M1=M
M2=MA+MF
XMU1=AMU(T1)
XMU2=AMU(TT)
XRHO1=ARHO(T1,P1)
XRHO2=ARHO(TT,P2)
B=(M1/M2)**(-1.75)*(XMU1/XMU2)**(-.25)*(XRHO1/XRHO2)*(P1/P2)
DPH1=DPH2*DHDHT**3/B
DPPCLR=DP-DPPWHB-DPPREG-DPH1
DPPREC=DPPH-DPH2-.1
RETURN
END

SUBROUTINE TURBVOL (G,M,V,TR,T,VOL)
REAL M,N,KI1,IT
SF=4.0
UT=850.
PHIT=0.6
VA=0.9
CL=2.0
PI=3.14159
RHOM=15.12
A=9.63E6
B=-95.14
ALPHA=1.2
A1=2.5
X1=.307
KI1=3.39E-3
THETA=0.8
RT=SQRT(M*V/(PHIT*UT*PI*(1.-VA**2)))
VB=SQRT(1.-TR**(1./G)*(1.-VA**2))
C1=0.2*(A1+1.)*1./V*(1.-VA**2)*UT**2*RT**2*X1/KI1
C2=2.*(A-B*T)/(SF*ALPHA)
C3=2.*B*T*(1.-VA**2)**(G-1.)/(SF*ALPHA)
C4=-1.*THETA*RHOM*UT**2/2.
H=(VA-VB)/20.
SUM=0.0
X1=0.0
Y1=VA
DO 100 J=1,21
  N=C1*((1.-Y1)/(1.+Y1))*PHIT**2+Y1**2
```

```
      D=C2+C3*(1.-Y1**2)**(1.-G)+C4*(1.-Y1**2)
      DN=C1*(2.*Y1*(1.-Y1)/(1.+Y1)-(PHIT**2+Y1**2)/(1.+Y1)-(1.+Y1)
     ***(-2)*(1.-Y1)*(PHIT**2+Y1**2))
      DD=2.*Y1*(1.-Y1**2)**(-1.*G)*(G-1.)*C3-2.*Y1*C4
      X2=.5*(1./(N/D)**.5)*((DN*D-DD*N)/D**2)
      IF(X1 .EQ. 0.0) GO TO 50
      SUM=SUM+(X1+X2)/2.
50    X1=X2
      Y1=Y1-H
100   CONTINUE
      IT=H*SUM
      VOL=2.*PI*RT**2*CL*IT
      RETURN
      END

      SUBROUTINE COMPVOL (G,M,V,PR,VOL)
      REAL M,N,KI1,IT
      SF=4.0
      UT=850.
      PHIT=0.5
      VA=0.6
      CL=2.0
      PI=3.14159
      RHOM=15.12
      ALPHA=1.2
      A1=12.0
      X1=.05
      KI1=4.93E-5
      THETA=0.8
      SIGMAY=12.54E6
      RT=SQRT(M*V/(PHIT*UT*PI*(1.-VA**2)))
      VB=SQRT(1.-PR**(-1./G)*(1.-VA**2))
      C1=.2*(A1+1.)*1./V*(1.-VA**2)*UT**2*RT**2*PHIT**2*X1/KI1
      C2=-1*THETA*RHOM*UT**2/2.
      C3=2.*SIGMAY/(SF*ALPHA)
      H=(VB-VA)/20.
      SUM=0.0
      X1=0.0
      Y1=VA
      DO 100 J=1,21
      N=C1*((1.-Y1)/(1.+Y1))*(PHIT**2+.49*Y1**2)/(PHIT**2+.7225*Y1**2
     *)
      D=C3+C2*(1.-Y1**2)
      DN=C1*(-1.*(PHIT**2+.49*Y1**2)/(PHIT**2+.7225*Y1**2)*(1.+Y1)-(1
     *.-Y1)*(1.+Y1)**(-2)*(PHIT**2+.49*Y1**2)/(PHIT**2+.7225*Y1**2)+
     *.98*Y1*(1.-Y1)/(1.+Y1)/(PHIT**2+.7225*Y1**2)-1.445*Y1*(1.-Y1)/
     *(1.+Y1)*(PHIT**2+.49*Y1**2)*(PHIT**2+.7225*Y1**2)**(-2))
      DD=-2.*C2*Y1
      X2=.5*(1./(N/D)**.5)*((DN*D-DD*N)/D**2)
      IF(X1 .EQ. 0.0) GO TO 50
      SUM=SUM+(X1+X2)/2.
```

```
50    X1=X2
      Y1=Y1+H
100   CONTINUE
      IT=-H*SUM
      VOL=2.*PI*RT**2*CL*IT
      RETURN
      END

      SUBROUTINE REGEN (T2,T3,T5,T6,P,D,MB,PR,DPPREG,DHDREG,VOL)
      REAL MB
      T=T5/2+T6/2
      P1=P*PR
      SIGMA=ASIGMA(T,P1)
      IF(T2 .EQ. T3) GO TO 99
      DTLM=T3-T2
      GAMMA=1./(6803.*P1)**.41*(.0254*D)**1.18/(.56*DTLM)**1.41
      DH1=CP2(T5)-CP2(T6)
      PI=3.14159
      AREG=PI/4.*SIGMA*GAMMA*(MB*.454)*(DH1*2326.)**1.41*(1.+DHDREG)
     ***2.41*(1.+PR**(+2)*DHDREG**(-3))**.41*35.3
      VOL=AREG*DPPREG**(-.41)
99    RETURN
      END

      SUBROUTINE RECUP (T2H,T3H,T5H,T6H,PRH,DPPH,D,MA,MF,DPPREC,VOL)
      REAL MA,MF
      T=T5H/2.+T6H/2.
      PC=1./(1.-DPPH)
      P=14.7*PRH
      SIGMA=ASIGMA(T,P)
      DTLM=T3H-T2H
      GAMMA=(6803.*P)**(-.41)*(.0254*D)**1.18*(.56*DTLM)**(-1.41)
      DH1=CP2(T5H)-CP2(T6H)
      PI=3.14159
      DHDREC=.65
      CM=MA+MF
      AREC=PI/4.*SIGMA*GAMMA*(CM*.454)*(DH1*2326.)**1.41*(1.+DHDREC)
     ***2.41*(1.+DHDREC**(-3)*PC**2)**.41*35.3
      VOL=AREC*DPPREC**(-.41)
      RETURN
      END

      SUBROUTINE COOLER (T7,T1,P,D,NTP,MB,DHDCLR,DPPCLR,VOL)
      REAL NPP,NTP,MB
      IF(T7 .LT. T1) GO TO 99
      TA=T7/2.+T1/2.
      SIGMA=ASIGMA(TA,P)
      TC1=530.
      TC2=540.
      DTLM=(T1-TC1-T7+TC2)/LOG((T1-TC1)/(T7-TC2))
      GAMMA=(6803.*P)**(-.41)*(.0254*D)**1.18*(.56*DTLM)**(-1.41)
```

```
      DH1=CP2(T7)-CP2(T1)
      PI=3.14159
      TW=TC1/2.+TC2/2.
      RK=AK(TA,P)/1.E-4
      RMU=AMU(TA)/6.4E-4
      RM=(T7-T1)*ACP(TA)/(1.*10.)
      RPR=APR(TA,P)/6.4
      NPP=.80
      RRHO=ARHO(TA,P)/64.0
      RSV=ARHO(T7,P)/64.0
      ALPHA=RK*RMU**(-.8)*RM**.8*RPR**.4
      CB=RM**(-2.75)*RMU**(-.25)*RRHO*RSV/NTP/NPP
      ACLR=PI/4.*SIGMA*GAMMA*(MB*.454)*(DH1*2326.)**1.41*(1.-DHDCLR)
     ***(1.+ALPHA*DHDCLR)**1.41*(1.+CB*DHDCLR**(-3))**.41*35.3
      VOL=ACLR*DPPCLR**(-.41)
   99 RETURN
      END

      SUBROUTINE COMBUSTOR (T3,T4,T4H,T5H,WP,PR,D,MB,PRH,MA,MF,VOL)
      REAL MB,MA,MF,M
      COMMON /COMB/DPPRG,DPPRC,DPPHE,DHDHR,DPHE2
      T=T3/2+T4/2
      P=WP*(PR-DPPRG)
      SIGMA=ASIGMA(T,P)
      DTLM=(T5H-T3-T4H+T4)/LOG((T5H-T3)/(T4H-T4))
      GAMMA=(6803.*P)**(-.41)*(.0254*D)**1.18*(.56*DTLM)**(-1.41)
      DH1=CP2(T4)-CP2(T3)
      PI=3.14159
      TH=T4H/2+T5H/2
      TRH=(1.-DPPHE)*PRH
      PH=14.7*(PRH-DPPRC)
      RK=AK(T,P)/AK(TH,PH)
      RMU=AMU(T)/AMU(TH)
      M=MA+MF
      RM=MB/M
      RPR=APR(T,P)/APR(TH,PH)
      ALPHA=RK*RMU**(-.8)*RM**.8*RPR**.4
      AHTR=PI/4.*SIGMA*GAMMA*(MB*.454)*(DH1*2326.)**1.41*(1.+DHDHR)*
     *  (1.+ALPHA*DHDHR)**1.41*DHDHR**(-1.23)*35.3
      VOL=AHTR*DPHE2**(-.41)
      RETURN
      END

      SUBROUTINE WHBVOL (MB,MS,T6,T7,NWHB,H2S,H3S,DPPWHB,VOL)
      REAL MB,MS,NWHB,NTU
      T=T6/2+T7/2
      CP=ACP(T)
      CPS=2.0
      CR=MS*CPS/(MB*CP)
      IF(CR .GT. 1.) GO TO 10
      CMIN=MS*CPS
```

```
      GO TO 20
10    CR=1./CR
      CMIN=CP*MB
20    PS=4.0
      ER=(NWHB*CR-1.)/(NWHB-1.)
      NTU=(-PS)*LOG(1.+1./CR*LOG(1.-CR*(1.-ER**(1./PS))/(CR-ER**(1./
     *PS))))
      DH=H3S-H2S
      VOL=.00045*CMIN*NTU*MS*DH*DPPWHB**(-.41)
      RETURN
      END

      SUBROUTINE CONDVOL (MS,X,VOL)
      REAL MS
      VOL=MS*X*27.3
      RETURN
      END
```

```
      SUBROUTINE STMTAB (NI,XI,TI)
      DIMENSION SATLIQ(3),STEAM(3),COEFFT(237)
      COMMON /STM/PRESS,TEMP,QUAL,SATLIQ,STEAM
      DATA (COEFFT(I),I=1,30)     / .22082221E+04, .55000000E-03
     *, .69573391E+03, .60464261E+01, .70000000E+01, .13552680E-00
     *,-.19513682E+00,-.52279952E-00, .91148958E-01, .51843682E-01
     *,-.50625849E+00, .72595720E+00, .14585694E+02, .12404319E-03
     *, .55592942E-03, .20000000E+01,-.74133440E-06, .73324823E-06
     *, .26548914E-04, .91349306E-04, .15679922E-02, .25032530E-01
     *, .30706919E+00, .36241343E+01, .44960616E-02, .17926732E+03
     *, .45000000E+03, .55000000E+03,-.98736888E-22, .19301230E-19/
      DATA (COEFFT(I),I= 31, 60)    / .30041189E-17,-.42113669E-15
     *,-.32893178E-14,-.11734466E-11, .11244476E-07,-.78419653E-05
     *, .81536841E-02, .69519265E-01, .25000000E+03, .20575548E-23
     *,-.64272696E-21, .12704366E-18,-.36976163E-16, .63370000E-13
     *,-.54251564E-10, .44797322E-07,-.28104756E-04, .17490541E-01
     *, .33953897E+01, .43010158E+03, .54927377E+03,-.18444779E-19
     *, .92756543E-17,-.95470656E-15,-.29004677E-12, .35353292E-10
     *, .39936598E-08,-.99567313E-06,-.59157992E-03, .78143274E-00/
      DATA (COEFFT(I),I= 61, 90)    / .55000000E+03, .21848036E+03
     *,-.14722587E-21, .56553573E-19, .47725605E-17,-.53349163E-14
     *, .27455505E-12,-.18335302E-10,-.31160333E-06,-.11956354E-03
     *, .98512754E+00, .25000000E+03, .47000000E+03, .55000000E+03
     *, .73857180E-22, .33225506E-19, .30075575E-18,-.73898651E-15
     *,-.70679233E-14, .90114201E-11, .58149642E-09, .36712981E-07
     *, .30734203E-04, .21762691E-01, .25000000E+03,-.78459129E-25
     *,-.33608032E-23, .95319792E-20, .25729365E-18,-.31591904E-15/
      DATA (COEFFT(I),I= 91,120)    / .60506309E-13, .13684599E-10
     *, .15587770E-07, .81264440E-05, .17002639E-01, .45000000E+03
     *, .55000000E+03, .94743147E-18, .24971478E-16,-.38302588E-14
     *, .44979672E-12,-.51337380E-10, .49178986E-08, .48326895E-05
     *, .12399614E-02, .12798551E+01, .54927377E+03, .25000000E+03
     *, .12945371E-21,-.65873139E-19,-.13817574E-17, .62520106E-14
     *, .17548001E-12, .23202424E-09, .36145967E-06, .12506461E-03
     *, .10150974E+01, .21848036E+03, .45000000E-03, .55000000E+03/
      DATA (COEFFT(I),I=121,150)    /-.21134438E-20, .15327038E-18
     *, .69217855E-16,-.18345822E-14,-.66417458E-12, .14589563E-10
     *, .59598410E-08, .45575915E-06, .12325099E-02, .74969797E+00
     *, .25000000E+03, .50084539E-23,-.15659070E-21,-.46099119E-18
     *, .49239634E-17, .16250351E-13,-.74068320E-12, .10479899E-08
     *,-.85546496E-06, .14285709E-02, .36753405E+00, .17000000E+01
     *, .19000000E+01,-.11085077E+05,-.15994587E+05, .91407688E+04
     *,-.18796109E+03, .15699797E+03,-.20213342E+03,-.48650176E+02/
      DATA (COEFFT(I),I=151,180)    / .15040455E+03,-.19050547E+03
     *, .11199490E+04, .15000000E+01,-.14863585E+06,-.13414819E+06
     *, .62379937E+05, .29474498E+05,-.18907586E+05,-.57858876E-03
     *, .35122466E+04,-.12340833E+04,-.68928827E+02, .12036836E+04
     *, .30000000E+01, .13000000E-04, .70848990E+01, .16507788E-02
     *, .52122736E+03,-.11682947E-01,-.12001588E+00, .20411991E+01
     *, .12340954E-04, .13555136E-03,-.15099319E-03, .30000000E-01
     *, .18000000E+01, .23822091E+02, .18776649E+03, .33080504E-03/
```

```
      DATA (COEFFT(I),I=181,210)    /-.32141408E+01, .42449101E-03
     *, .16400835E+04, .18746657E+02, .53978873E+02, .20719841E-04
     *, .13000000E+04, .13000000E+01,-.64369800E-05, .96418803E-02
     *, .38879248E+01, .11034060E-05, .47225001E-03,-.94113600E-01
     *, .69416973E-05,-.60892120E-02, .10951922E+01, .13000000E-04
     *, .18000000E+01,-.33101208E-03, .21661916E+01, .53600291E+03
     *,-.77236618E-03, .36753650E+00,-.25450121E+03, .25249707E-02
     *,-.16427106E+01, .61477766E+03,      0.0,           0.0     /
      DATA (COEFFT(I),I=211,237)    /      0.0, 0.0, 0.0, 0.0, 0.0
     *, 0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0,0.0, .90000000E-03
     *, .14000000E+01,-.15478259E-04, .19351556E-01, .24698909E-01
     *, .23329644E-04, .11493791E-02,-.13139208E+02, .52860219E-05
     *,-.12592899E-01, .55101160E+01/
      N=NI
      X=XI
      Y=YI
      I=N-3
      LINK=0
      PRESS=X
      DELTA1=0.0
      ERROR1=0.0
      ERROR2=0.0
      IF (N-6) 1000,1000,1030
 1000 IF (I) 1020,1020,1010
 1010 LINK=1
      N=I
 1020 IF (X-COEFFT(N)) 1040,1040,1620
 1030 N=I
 1040 GO TO (1130,1060,1190,1230,1080,1090,1180,1330),N
 1050 X=Z
      N=2
 1060 TEMP=X
      GO TO 1190
 1070 PRESS=EXP(Z)
      GO TO 1150
 1080 II=2
      JJ=3
      TOLER=.005
      GO TO 1100
 1090 II=3
      JJ=2
      TOLER=.000005
 1100 H=Y
 1110 LINK=2
 1120 N=1
 1130 X=ALOG(PRESS)
      PLOG=X
      GO TO 1190
 1140 TEMP=Z
 1150 X=TEMP
      J=1
```

```
      N=4
      GO TO 1190
1160  SATLIQ(J)=Z
      J=J+1
      N=N+1
      IF (J-3) 1190,1190,1170
1170  IF (LINK) 1640,1640,1300
1180  H=X
      S=Y
      X=S
1190  I=23*N-19
      K=1
      IF (X-COEFFT(I)) 1200,1200,1210
1200  K=12
1210  I=I+K
      ARG=X-COEFFT(I)
      Z=0.0
      DO 1220 K=1,10
      I=I+1
1220  Z=Z*ARG+COEFFT(I)
      GO TO (1140,1070,1050,1160,1160,1160,1410),N
1230  TEMP=Y
      GO TO 1300
1240  ERROR=H-STEAM(II)
      IF (ABS(ERROR)-TOLER) 1640,1250,1250
1250  IF (ERROR1) 1260,1280,1260
1260  ERROR1=ERROR1-ERROR
      IF (ABS(ERROR1)-TOLER) 1290,1290,1270
1270  SLOPE=DELTA1/ERROR1
1280  DELTA1=SLOPE*ERROR
      ERROR1=ERROR
1290  TEMP=TEMP+DELTA1
1300  IF (PRESS-COEFFT(1)) 1320,1320,1310
1310  IF ((.00001178*PRESS+.09411)*PRESS+382.1-TEMP) 1320,1320,1620
1320  T=.55555556*TEMP+255.38223
      TLOG=ALOG(T)
      TAU=1.0/T
      COEFFT(222)=TAU*TAU*186210.06
      COEFFT(217)=EXP(COEFFT(222)+7.8791476-TLOG)
      COEFFT(221)=162460.*TAU
      COEFFT(220)=125970.*TAU
      COEFFT(219)=EXP(149.27765-23.*TLOG)
      COEFFT(212)=1.89-COEFFT(217)
      COEFFT(211)=82.546-COEFFT(221)
      COEFFT(210)=.21828*T-COEFFT(220)
      COEFFT(209)=COEFFT(219)-.0003635*T
      COEFFT(222)=(COEFFT(222)+COEFFT(222)+1.0)*COEFFT(217)
      COEFFT(217)=COEFFT(212)-COEFFT(222)
      COEFFT(214)=COEFFT(217)/COEFFT(212)
      COEFFT(216)=COEFFT(211)*COEFFT(214)+41.273-COEFFT(221)
      COEFFT(215)=COEFFT(210)*COEFFT(214)-.5*COEFFT(220)
```

```
      COEFFT(214)=COEFFT(209)*COEFFT(214)+1.8461538*COEFFT(219)
      COEFFT(221)=.50*COEFFT(211)-COEFFT(216)
      COEFFT(220)=.25*COEFFT(210)-COEFFT(215)
      COEFFT(219)=.076923077*COEFFT(209)-COEFFT(214)
      COEFFT(218)=(.0037299965*T+14.531813)*T+17806.513+472.2493*TLOG
      COEFFT(223)=-.0074599931*T-14.531813*TLOG+2.4524439+472.24937
     **TAU
 1330 P=.068046190*PRESS
      PLOG=ALOG(PRESS)
      COEFFT(213)=COEFFT(212)*TAU
      COEFFT(226)=COEFFT(213)*P
      COEFFT(225)=COEFFT(226)*COEFFT(226)
      COEFFT(224)=COEFFT(225)*COEFFT(225)
      COEFFT(224)=COEFFT(224)*COEFFT(224)*COEFFT(226)
      COEFFT(226)=COEFFT(226)*COEFFT(213)
      COEFFT(213)=P*TAU
      STEAM(1)=(((COEFFT(209)*COEFFT(224)+COEFFT(210))*COEFFT(225)+
     *COEFFT(211))*COEFFT(226)+COEFFT(212)+4.55504/COEFFT(213))
     **.0160185
      STEAM(2)=((((COEFFT(214)*COEFFT(224)+COEFFT(215))*COEFFT(225)+
     *COEFFT(216))*COEFFT(226)+COEFFT(217))*P+COEFFT(218))*.04355685
      STEAM(3)=((((COEFFT(219)*COEFFT(224)+COEFFT(220))*COEFFT(225)+
     *COEFFT(221))*COEFFT(226)+COEFFT(222))*COEFFT(213)+COEFFT(223)
     *+4.55504*PLOG)*(-.02419825)
      QUAL=1.0
      I=LINK+1
      GO TO (1640,1640,1340,1240,1340,1450),I
 1340 X=STEAM(II)
      IF (H-X) 1350,1640,1360
 1350 Y=SATLIQ(II)
      QUAL=(H-Y)/(X-Y)
      X=1.0-QUAL
      STEAM(1)=QUAL*STEAM(1)+X*SATLIQ(1)
      STEAM(JJ)=QUAL*STEAM(JJ)+X*SATLIQ(JJ)
      STEAM(II)=H
      IF (LINK-2) 1640,1640,1460
 1360 I=(II-1)*11+154
      J=1
      X=PLOG
 1370 DO 1380 K=209,219
        COEFFT(K)=COEFFT(I)
 1380   I=I+1
      X=X-COEFFT(209)
      Y=Y-COEFFT(210)
      COEFFT(209)=(COEFFT(211)*X+COEFFT(212))*X+COEFFT(213)
      COEFFT(210)=(COEFFT(214)*X+COEFFT(215))*X+COEFFT(216)
      COEFFT(211)=((COEFFT(217)*X+COEFFT(218))*X+COEFFT(219))*Y
      Z=COEFFT(211)+COEFFT(210)
      SLOPE=COEFFT(211)+Z
      Z=Z*Y+COEFFT(209)
      GO TO (1390,1430,1530,1540),J
```

```
1390 LINK=3
     IF (Z-TEMP) 1300,1300,1400
1400 TEMP=Z
     GO TO 1300
1410 J=2
     I=227
     X=H
     IF (Z-H) 1420,1370,1370
1420 I=187
     GO TO 1370
1430 DPDS=SLOPE
     PRESS=EXP(Z)
     PLOG=Z
     IF (I-227) 1520,1520,1440
1440 LINK=4
     II=2
     JJ=3
     GO TO 1120
1450 DH=H-STEAM(2)
     STEAM(3)=STEAM(3)+DH/(TEMP+459.69)
1460 DS=S-STEAM(3)
     DELTA1=STEAM(3)-ERROR1
     IF (ERROR1) 1470,1490,1470
1470 IF (ABS(DELTA1)-.000005) 1500,1500,1480
1480 DPDS=(PLOG-PLOG1)/DELTA1
1490 PLOG1=PLOG
     ERROR1=STEAM(3)
1500 PRESS=PRESS*(1.0+DS*DPDS)
     IF (LINK-4) 1510,1510,1550
1510 IF (ABS(DS)-.000005) 1140,1120,1120
1520 LINK=5
     X=H
     Y=S
     I=198
     J=3
     GO TO 1370
1530 TEMP=Z
     DTDS=SLOPE
     X=PLOG
     Y=H
     I=165
     J=4
     GO TO 1370
1540 DTDH=SLOPE
     GO TO 1300
1550 TEMP=TEMP+DS*DTDS
     IF (ABS(DH)-.005) 1610,1560,1560
1560 DELTA2=DH-ERROR2
     IF (ERROR2) 1570,1590,1570
1570 IF (ABS(DELTA2)-.005) 1600,1600,1580
1580 DTDH=(TEMP1-TEMP)/DELTA2
```

```
1590 TEMP1=TEMP
     ERROR2=DH
1600 TEMP=TEMP+DH*DTDH
     GO TO 1320
1610 IF (ABS(DS)-.000005) 1640,1600,1600
1620 WRITE (6,1630) NI,XI,YI
1630 FORMAT (I3,2E14.3,13H OUT OF RANGE)
     STOP
1640 RETURN
     END
```

APPENDIX B

BRAYTON CYCLE: PRESSURE RATIO=2.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS FLOW (Lbm/s)=444.37

STATE	TEMP(deg R)
1	540.0
2	675.7
3	1668.3
4	2060.0
5	1304.2
6	811.2
PP	784.0
7	778.3

SUMMARY: THERMAL EFF=0.370
HEATER EFF=0.93[illegible] (PHI=0.90)
SFC(Lbm/HP-hr)=0.572
VOL TOTAL(cu ft)=516.5
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.045

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.078	.744	15.5	EFF=0.88
TURBINE			10.0	EFF(P)=0.91
COOLER	.007	.260	139.8	
COMPRESSOR			29.3	EFF(P)=0.88
HEATER	.0005			
BOILER	.014			EFF=0.94

HEATER CYCLE: PRESSURE RATIO = 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)=47.84
FUEL FLOW(Lbm/s)= 3.97

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1727.4
4h	3054.5
5h	1768.8
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.321	.650	2.6	EFF=0.96
TURBINE			1.1	
HEATER	.012	.310	67.4	
COMPRESSOR			16.0	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 56.7
STEAM FLOW(Lbm/s)= 2.96
FLOW RATIO=0.0067

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	56.7	
SF	749.0	56.7	
3s	776.2	56.7	
4s	540.0	0.5	QUAL=0.844

COMPONENT	VOL(cu ft)	
BOILER	162.3	
TURBINE	4.3	EFF(IS)=0.88
CONDENSER	68.3	

BRAYTON CYCLE: PRESSURE RATIO= 3.00
WORKING PRESSURE =135.0
TOTAL PRESSURE DROP=.10
MASS FLOW(Lbm/s)=287.63

SUMMARY:
- THERMAL EFF=0.432
- HEATER EFF=0.933 (PHI=0.90)
- SFC(Lbm/HP-hr)=0.490
- VOL TOTAL(cu ft)= 318.1
- POWER(HP)=25000.0
- WORK FRAC BY STM CYCLE=0.04[illegible]

STATE	TEMP(deg R)
1	540.0
2	769.8
3	1538.8
4	2060.0
5	1643.6
6	874.7
PP	841.2
7	831.8

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.050	.837	11.5	EFF=0.88
TURBINE			5.3	EFF(P)=0.91
COOLER	.024	.352	88.9	
COMPRESSOR			20.9	EFF(P)=0.88
HEATER	.0004			
BOILER	.026			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)= 40.99
FUEL FLOW(Lbm/s)= 3.40

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1586.5
4h	4932.9
5h	1638.8
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.310	.650	2.2	EFF=0.93
TURBINE			0.9	EFF(P)=0.89
HEATER	.022	.358	28.6	
COMPRESSOR			12.7	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 128.0
STEAM FLOW(Lbm/s)= 2.48
FLOW RATIO=0.0086

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	128.0	
SF	806.2	128.0	
3s	839.7	128.0	
4s	540.0	0.5	QUAL=0.819

COMPONENT	VOL(cu ft)	
BOILER	88.3	
TURBINE	3.4	EFF(IS)=0.88
CONDENSER	55.5	

BRAYTON CYCLE: PRESSURE RATIO= 4.00
WORKING PRESSURE = 135.0
TOTAL PRESSURE DROP=.10
MASS FLOW (Lbm/s)=239.91

STATE	TEMP(deg R)
1	540.0
2	843.9
3	1453.8
4	2060.0
5	1536.9
6	927.0
PP	876.3
7	860.6

SUMMARY: THERMAL EFF=0.447
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.473
VOL TOTAL(cu ft)= 352.2
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.063

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.021	.930	13.2	EFF=0.88
TURBINE			3.6	EFF(P)=0.91
COOLER	.032	.444	39.8	
COMPRESSOR			18.4	EFF(P)=0.88
HEATER	.0003			
BOILER	.046			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)=39.59
FUEL FLOW(Lbm/s)= 3.29

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1494.5
4h	4855.0
5h	1553.8
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.299	.650	2.1	EFF=0.92
TURBINE			0.8	EFF(P)=0.89
HEATER	.033	.406	17.7	
COMPRESSOR			12.0	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 199.9
STEAM FLOW(Lbm/s)= 3.19
FLOW RATIO=0.0133

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	199.9	
SF	841.8	199.9	
3s	892.0	199.9	
4s	540.0	0.5	QUAL=0.810

COMPONENT	VOL(cu ft)	
BOILER	119.0	
TURBINE	5.1	EFF(IS)=0.88
CONDENSER	70.5	

BRAYTON CYCLE: PRESSURE RATIO= 5.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS FLOW (Lbm/s)=212.34

STATE	TEMP(deg R)
1	540.0
2	905.8
3	1391.3
4	2060.0
5	1458.1
6	972.1
PP	876.8
7	846.9

SUMMARY: THERMAL EFF=0.460
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.460
VOL TOTAL(cu ft)=596.1
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.111

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.010	.950	15.7	EFF=0.88
TURBINE			2.7	EFF(P)=0.91
COOLER	.022	.536	107.3	
COMPRESSOR			16.7	EFF(P)=0.88
HEATER	.0003			
BOILER	.068			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)= 38.49
FUEL FLOW(Lbm/s)= 3.19

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1427.4
4h	4798.9
5h	1491.8
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.289	.650	2.1	EFF=0.90
TURBINE			0.8	EFF(P)=0.89
HEATER	.044	.454	12.4	
COMPRESSOR			11.5	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 200.0
STEAM FLOW(Lbm/s)= 5.21
FLOW RATIO=0.0245

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	200.0	
SF	841.8	200.0	
3s	937.1	200.0	
4s	540.0	0.5	QUAL=0.826

COMPONENT	VOL(cu ft)	
BOILER	298.1	
TURBINE	10.9	EFF(IS)=0.88
CONDENSER	117.4	

BRAYTON CYCLE: PRESSURE RATIO = 6.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS FLOW(Lbm/s)=197.00

STATE	TEMP(deg R)
1	540.0
2	959.4
3	1343.8
4	2060.0
5	1396.2
6	1011.8
PP	376.8
7	335.3

SUMMARY: THERMAL EFF=0.464
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.456
VOL TOTAL(cu ft)=879.0
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.151

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.010	.950	14.4	EFF=0.86
TURBINE			2.2	EFF(P)=0.91
COOLER	.016	.628	129.7	
COMPRESSOR			15.8	EFF(P)=0.86
HEATER	.0003			
BOILER	.074			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)=38.17
FUEL FLOW(Lbm/s)- 3.17

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1375.4
4h	4755.8
5h	1443.8
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.278	.650	2.1	EFF=0.89
TURBINE			0.8	EFF(P)=0.89
HEATER	.055	.502	9.5	
COMPRESSOR			11.4	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 200.0
STEAM FLOW(Lbm/s)= 6.70
FLOW RATIO=0.0340

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	200.0	
SF	841.8	200.0	
3s	976.8	200.0	
4s	540.0	0.5	QUAL=0.838

COMPONENT	VOL(cu ft)	
BOILER	523.6	
TURBINE	16.2	EFF(IS)=0.88
CONDENSER	153.3	

```
BRAYTON CYCLE:  PRESSURE RATIO=7.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS Flow(Lbm/s)=187.48
     STATE    TEMP(deg R)    SUMMARY:THERMAL   EFF=0.463
       1        540.0                HEATER   EFF=0.933 (PHI=0.90)
       2       1007.0                SFC(Lbm/HP=hr)=0.457
       3       1304.9                VOL TOTAL(cu ft)=1195.7
       4       2060.0                POWER(HP)=25000.0
       5       1345.5                WORK FRAC BY STM CYCLE=0.189
       6       1047.6
       PP       876.8
       7        825.2
   COMPONENT   DPP   DHD   VOL(cu ft)
     REGEN     .010  .950      13.6  EFF=0.88
    TURBINE                     1.9  EFF(P)=0.91
    COOLER     .012  .720     155.9
  COMPRESSOR                   15.3  EFF(P)=0.88
    HEATER     .0004
    BOILER     .078                  EFF=0.94
HEATER CYCLE:  PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)= 38.22
FUEL FLOW(Lbm/s)= 3.17
     STATE    TEMP(deg R)
       1h       535.0
       2h       835.2
       3h      1333.3
       4h      4721.2
       5h      1404.9
       6h       945.0
       7h       760.0
   COMPONENT   DPP   DHD   VOL(cu ft)
     RECUP     .267  .650      2.1   EFF=0.87
    TURBINE                    0.8   EFF(P)=0.89
    HEATER     .066  .550      7.7
  COMPRESSOR                  11.4   EFF(P)=0.85
RANKINE CYCLE:  SATURATION PRESSURE= 200.0
STEAM FLOW(Lbm/s)= 7.92
FLOW RATIO=0.0422
     STATE    TEMP(deg R)    PRESSURE(psi)
       1s       540.0            0.5
       2s       555.0          200.0
       SF       841.8          200.0
       3s      1012.6          200.0
       4s       540.0            0.5  QUAL=0.849
   COMPONENT   VOL(cu ft)
    BOILER       782.4
    TURBINE       21.0  EFF(IS)=0.88
  CONDENSER      183.6
```

BRAYTON CYCLE: PRESSURE RATIO= 8.00
WORKING PRESSURE= 135.0
TOTAL PRESSURE DROP= .10
MASS FLOW(Lbm/s) = 131.22

STATE	TEMP(deg R)
1	540.0
2	1049.8
3	1272.5
4	2060.0
5	1302.9
6	1080.2
PP	876.8
7	816.3

SUMMARY: THERMAL EFF=0.460
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.459
VOL TOTAL(cu ft)=1550.7
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.225

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.010	.950	13.1	EFF=0.88
TURBINE			1.6	EFF(P)=0.91
COOLER	.010	.812	181.1	
COMPRESSOR			15.1	EFF(P)=0.88
HEATER	.0004			
BOILER	.080			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)=38.45
FUEL FLOW(Lbm/s)= 3.19

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1298.2
4h	4692.5
5h	1372.5
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.256	.650	2.1	EFF=0.86
TURBINE			0.8	EFF(P)=0.89
HEATER	.076	.598	6.5	
COMPRESSOR			11.5	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE=200.0
STEAM FLOW(Lbm/s)= 8.97
FLOW RATIO=0.0495

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	200.0	
SF	841.8	200.0	
3s	1045.2	200.0	
4s	540.0	0.5	QUAL=0.859

COMPONENT	VOL(cu ft)	
BOILER	1083.1	
TURBINE	25.5	EFF(IS)=0.88
CONDENSER	210.2	

BRAYTON CYCLE: PRESSURE RATIO= 9.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS FLOW(Lbm/s)=176.95

SUMMARY: THERMAL EFF=0.456
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.468
VOL TOTAL(cu ft)=1950.3
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.260

STATE	TEMP(deg R)
1	540.0
2	1089.0
3	1245.0
4	2060.0
5	1266.2
6	1110.2
IP	876.8
7	808.2

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.010	.950	12.7	EFF=0.88
TURBINE			1.4	EFF(P)=0.91
COOLER	.010	.904	197.5	
COMPRESSOR			15.0	EFF(P)=0.88
HEATER	.0004			
BOILER	.080			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP = .43
AIR FLOW(Lbm/s)= 38.80
FUEL FLOW(Lbm/s)= 3.22

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1268.3
4h	4668.2
5h	1345.0
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.245	.650	2.2	EFF=0.85
TURBINE			0.8	EFF(P)=0.89
HEATER	.087	.646	5.7	
COMPRESSOR			11.7	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 200.00
STEAM FLOW(Lbm/s)= 9.90
FLOW RATIO=0.0560

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	200.0	
SF	841.8	200.0	
3s	1075.2	200.0	
4s	540.0	0.5	QUAL=0.867

COMPONENT	VOL(cu ft)	
BOILER	1439.0	
TURBINE	29.8	EFF(IS) =0.88
CONDENSER	234.5	

BRAYTON CYCLE: PRESSURE RATIO=10.00
WORKING PRESSURE=135.0
TOTAL PRESSURE DROP=.10
MASS FLOW (Lbm/s)=174.00

SUMMARY: THERMAL EFF=0.451
HEATER EFF=0.933 (PHI=0.90)
SFC(Lbm/HP-hr)=0.469
VOL TOTAL(cu ft)=2403.0
POWER(HP)=25000.0
WORK FRAC BY STM CYCLE=0.296

STATE	TEMP(deg R)
1	540.0
2	1125.1
3	1221.1
4	2060.0
5	1234.2
6	1138.1
PP	876.7
7	800.9

COMPONENT	DPP	DHD	VOL(cu ft)	
REGEN	.010	.950	12.4	EFF=0.88
TURBINE			1.3	EFF(P)=0.91
COOLER	.008	.996	232.8	
COMPRESSOR			14.9	EFF(P)=0.88
HEATER	.0004			
BOILER	.082			EFF=0.94

HEATER CYCLE: PRESSURE RATIO= 4.79
TOTAL PRESSURE DROP=.43
AIR FLOW(Lbm/s)= 39.22
FUEL FLOW(Lbm/s)= 3.25

STATE	TEMP(deg R)
1h	535.0
2h	835.2
3h	1242.5
4h	4647.3
5h	1321.1
6h	945.0
7h	760.0

COMPONENT	DPP	DHD	VOL(cu ft)	
RECUP	.235	.650	2.2	EFF=0.84
TURBINE			0.8	EFF(P)=0.89
HEATER	.098	.694	5.0	
COMPRESSOR			11.9	EFF(P)=0.85

RANKINE CYCLE: SATURATION PRESSURE= 199.9
STEAM FLOW(Lbm/s)= 10.76
FLOW RATIO=0.0618

STATE	TEMP(deg R)	PRESSURE(psi)	
1s	540.0	0.5	
2s	555.0	199.9	
SF	841.7	199.9	
3s	1103.1	199.9	
4s	540.0	0.5	QUAL=0.875

COMPONENT	VOL(cu ft)	
BOILER	1830.6	
TURBINE	34.0	EFF(IS)=0.88
CONDENSER	257.1	

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8